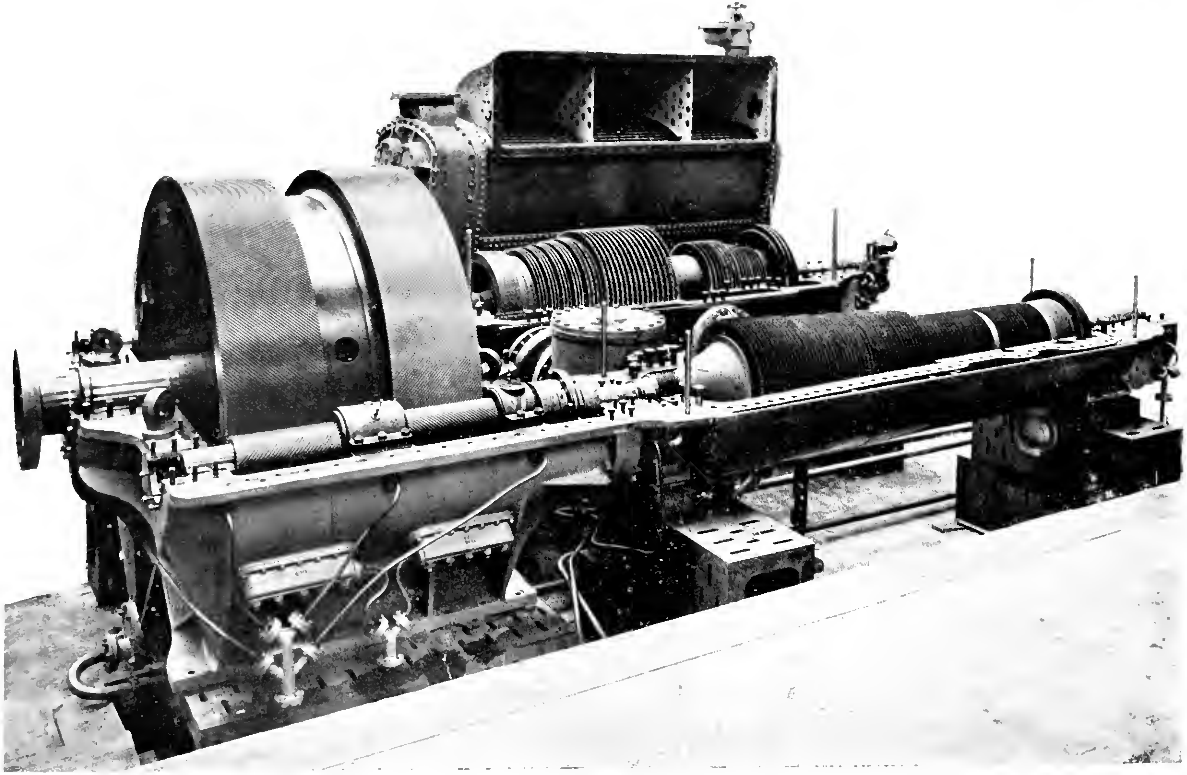
A MANUAL

OF

MARINE ENGINEERING.

[*Frontispiece.*



Turbines and Gearing of One Line of Propeller Shafts of Twin S.S. "Tuscania," 10,900 S.H.P. 17·65 Knots.
Spur Wheels, 10 feet diameter; Teeth, each 30 inches long; Revolutions, 137 per minute.
Pinions, 11 inches diameter; Revolutions, 1,500 per minute.
By Messrs. A. Stephen & Sons.

A MANUAL

OF

MARINE ENGINEERING:

COMPRISING

THE DESIGN, CONSTRUCTION, AND WORKING OF MARINE MACHINERY.

BY

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FORMERLY LECTURER ON MARINE ENGINEERING TO THE ROYAL NAVAL COLLEGE, GREENWICH; MEMBER OF THE INSTITUTION OF CIVIL ENGINEERS; MEMBER OF THE INSTITUTION OF NAVAL ARCHITECTS; MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS; MEMBER OF THE NORTH-EAST COAST INSTITUTION OF SHIPBUILDERS AND ENGINEERS; MEMBER OF THE INSTITUTE OF METALS, ETC.

With Numerous Tables and Illustrations reduced from Working Drawings and Photographs.

EIGHTEENTH EDITION, THOROUGHLY REVISED, GREATLY ENLARGED, AND MOSTLY RE-WRITTEN TO DATE.

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1918.

PREFACE TO THE EIGHTEENTH EDITION.

THE demand for this new Edition came when the War, with all its pomps and circumstances, has made it difficult to produce—both to Author and Publishers. Since the issue of the last Edition the changes in engineering practice have been many and great In some measure this is due to the special demands arising out of the War conditions, but it is largely due to the advance that goes on now, day by day, from the better knowledge of science gained by diligent research, and by the better application of it, whereby that experience is gained which engenders confidence as well as stimulates invention, and this produces improvements of many kinds.

The economic side of engineering, however, is asserting itself to a degree that never obtained in pre-War times, and it will surely remain as a predominating factor in all our every-day calculations for many years to come, so that we are compelled now to approach and to determine problems on lines not contemplated formerly. To be successful it will be necessary to cast aside all prejudices, to treat lightly the precedents, and to concentrate the solving of them—each on its own merits—by giving full heed to the physical and economic conditions only.

Necessity has ever been the mother of invention. To-day it will be likewise the remover of prejudice as well as the *alma* *mater* of research to all her children, so that they may thrive in a way they never have done hitherto in this country.

D. O. R. A. exercises a powerful influence over authors and publishers, whereby they are restrained from making public any of the wonderful advances achieved during the past four years, or to allude to the inventions whereby so much has been accomplished on the sea by the genius that otherwise might have remained dormant. Nevertheless this Edition does contain much that is new, and what was old has been renovated and brought up to date.

Special Appendices have been added which deal with the Heavy Oil Engine, Geared Turbines and Superheaters, as now in general use, and altogether the attempt is made to maintain the character of the Manual as far as circumstances will permit.

A. E. SEATON.

WESTMINSTER, *September*, 1918.

ORIGINAL PREFACE.

THE following Work has been prepared to supply the existing want of a Manual showing the application of Theoretical Principles to the Design and Construction of Marine Machinery, as determined by the experience of leading engineers, and carried out in the most recent successful practice. The data on which it is based, now first thrown into form for publication, have been collected during many years of study and practical work. It is hoped that the volume will be found useful by the engineer and draughtsman engaged in practice as a Handbook of Reference, and by the student, launched for the first time on the intricacies of Marine Construction, as a guide, supplying to some extent his lack of experience.

The rules and formulæ introduced (which have been divested as far as possible of complexity, and given in the simplest form attainable) may be used by any one who designs with some regard to theory, and, by varying the constants, be made to suit his own ideas of strength and stiffness. It may, perhaps, be thought by some that in certain instances details have been entered into with unnecessary minuteness; but it should be remembered, on the other hand, that not every engineer has the contents of a well-filled drawing-office to fall back upon in cases of doubt and difficulty.

It is hardly necessary to premise that it is wholly impossible to reconcile the practice of the naval designer, who thinks more of efficiency and weight than of cost, with that of the mercantile engineer, who studies efficiency and cost with but small regard to weight, and, therefore, few rules can be given which shall absolutely suit both. However, the manufacturer of machinery for the Merchant Service might follow with advantage much that has been proved to be good in naval practice, and the Naval Authorities might again, on their part, borrow from the Mercantile Marine a few suggestions which would render a warship, while no less efficient than at present, perhaps somewhat less intricate for those who have to work her.

In conclusion, the author can but express a hope that the publication of these notes, imperfect as they necessarily are, may tend to make a little clearer some of the technicalities of Marine Design and Construction, and so help forward, in however slight a degree, the application of scientific investigation to those problems which the marine engineer is called upon, day by day, to solve.

HULL, *January 30th*, 1883.

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A MANUAL

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MARINE ENGINEERING.

CHAPTER I.

GENERAL INTRODUCTION.

Fundamental Principles.—The first object aimed at by the marine engineer, is to propel a floating body through the water at a certain speed; the second, so to construct the propelling apparatus that the motion may readily be reversed; and the third, to adopt such an arrangement of propeller and engine as shall be convenient for the floating body and the service on which it is employed.

The principle on which nearly all marine propellers work is the projection of a mass of water in the direction opposite to that of the required motion. The only exception to this rule is the case of ferry steamers and some river craft, where a chain or rope lying in the bed of the river passes over a wheel or barrel in the ship itself.

The water, in modern practice, is projected by—(1) One or more screws at the end of the ship (as will be described under the heading of *propellers*); (2) by one or more paddle-wheels outside of the ship; or (3), by a form of wheel in the inside of the ship, which is generally spoken of as a *jet*-propeller, as the water issues in jets from orifices in the ship's side.

The latter, however, is seldom used, although it has some features that make it attractive in particular cases. The paddle-wheel, too, is slowly dying out; and although there is reason to believe that for river service, especially in the tropics, and certain special duties, it will survive, it is, nevertheless, gradually being displaced by the screw in some of these, while in all other services, even in the shallow water ones which at one time seemed reserved for its use, it is practically gone.

The paddle-wheel with feathering floats has certain qualities of its own, which render it more serviceable than the screw in particular cases; for example:—

In tug boats quick manœuvring is of extreme importance. The sudden stopping, and equally sudden and certain starting, of the boat is most desirable; this cannot be done with the screw, nor can the turning round in confined spaces be accomplished even with twin screws so dexterously as with disconnected paddle-wheels.

In river steamers running in very shallow water the paddle-wheel, especially when fitted at the stern, has distinct advantages over the screw, even when the latter is placed in a well or enclosure, as done by Mr. Yarrow and Sir John Thornycroft, inasmuch as it can free itself or be easily freed of weeds, is less liable to injury, and when injured is easily repaired.

In steamers making frequent and short calls at piers and wharfs, the paddle-wheel permits of a more rapid service on rough and windy days than can screws, especially the small screws of the turbine-driven ship.

On the other hand, the paddle-wheel is a heavy and somewhat clumsy instrument exposed to wind and sea, and so liable to damage, even when protected with boxes and guarded with sponsons, fenders, etc.; it can, however, be repaired by simple means, and even when badly damaged can

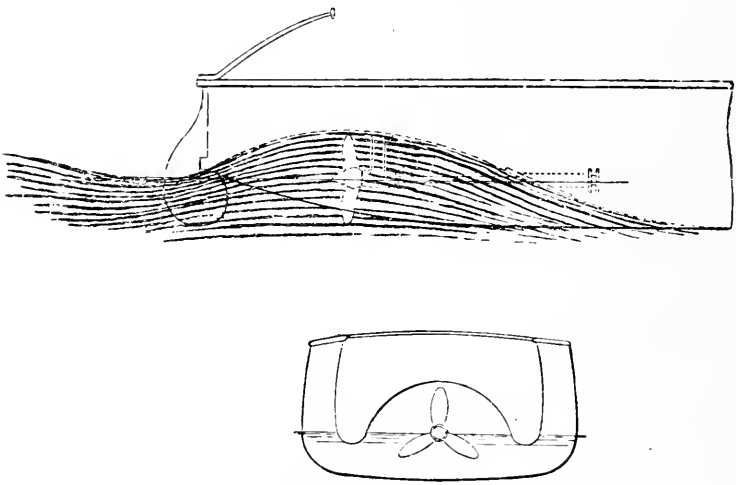


Fig. 1.—Thornycroft's Stern for Shallow Draught Screw Ships.

be generally sufficiently repaired by the ship's staff to permit of proceeding on the voyage.

Its position in the ship and the space occupied by the paddle engine interferes with the general economy; the machinery is heavier and more cumbersome than that driving a screw developing equal power, inasmuch as the speed of revolution is necessarily restricted; the wheel exerts a thrust on a part of the ship less calculated to take pressure than that to which the screw applies; but with the modern steel ship, however, that can be remedied, and provided for in a way which was not so easy with the wooden ship of the past.

The paddle-wheel was the propeller of the first steamers put to practical use, and to-day its efficiency is little short of that of the best screws.

The modern paddle engines, with their long stroke and choice design, have brought this about to a very great extent.

The screw is much smaller than even the small high-revolution paddle-wheel of to-day; it is wholly immersed and largely protected by the quarters of the ship. It is generally of so small a diameter as to be wholly below the water line, but in shallow draft ships this is not always possible; it is, however, totally immersed when working by surrounding it with portions of the body of the ship, so that it revolves in a channel, as done by Sir John Thornycroft (fig. 1), or by dropping a flap behind it, as done by Mr. Yarrow (fig. 2). The thrust from the screw is applied lower down, and nearer the centre line of the ship's resistance, than is that of a paddle-wheel, so that the tipping moment is quite small, and the thrust block is attached to a part of the ship's structure eminently calculated to take the force.

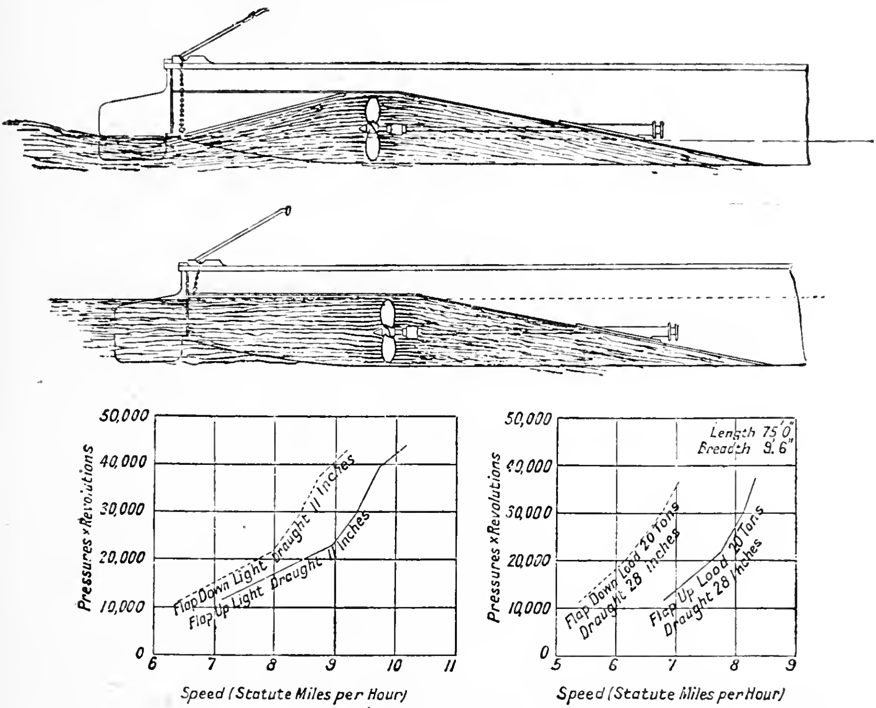


Fig. 2.—Yarrow's Drop Flap for Shallow Draught Screw Ships.

There being no restriction to the number of revolutions of the screw, the engines can thereby have a high piston speed, and be comparatively light, cheap, and made to occupy less space, and moreover can be placed in positions more convenient to the general arrangements of the ship. It used to be urged that the screw caused much vibration and consequent discomfort to those in the ship; that it was inherent in the screw, however, was erroneous, and it is known now that vibration was as much due to the momentum of the moving parts of the engines as to the screw; and, while it is admitted that even a good screw may cause horizontal vibration at the stern, owing to the difference in pressure of water on the lower and upper blades, it is only with the two-bladed variety that it is pronounced, and even then only with those of large diameter and small submersion is it excessive. With well designed three- and four-bladed screws driven at fairly high revolution by engines carefully balanced and well preserved, the vibration is virtually *nil*. That the screw is liable to foul itself with ropes, nets, etc., which only a diver can remove, and that it may damage itself by striking dock walls, semi-submerged wreckage, etc., so as to necessitate dry docking, or tipping must be admitted; but this seldom happens.

Hydromotors.—There is, however, another method by which ships have been, and may be, propelled distinguished from all others by the absence of any propelling instrument. In the past many proposals have been made and patented for ejecting a stream of water from the stern of a ship which has been taken in at the bow or through the bottom.

As far back at 1729, John Allen proposed this method, and in 1793 James Ramsey constructed and tried on the Thames a boat fitted with a pump and pipes to do this: the speed attained, however, was only 4 knots per hour. It was not, however, till 1849 that John Ruthven patented what has since been known as the jet propeller, and associated with his name, although he was not the first inventor of the system. In this case, however, the impeller of the centrifugal pump is really the propeller. In the case of the s.s. "Hydromotor," built and equipped in Holland in 1876, the stream was ejected from a pair of cylindrical receivers by steam pressing on the surface of the water which had been caused to flow into them by reason of the vacuum formed by condensing the steam of the previous discharge; in fact, they were a pair of pulsometers. The enterprise was a failure, and no one is likely ever again to attempt to propel a sea-going ship by such extravagant means—extravagant in steam consumption and in space occupied both by receivers and water passages.

Motive Power.—Both the screw and the paddle-wheel revolve around their axes, consequently the engine employed to move them must have circular motion, and inasmuch as both instruments are reversible, and can thereby reverse the motion of the ship, the engines should be capable of reversing. The engine must also be able to work efficiently at all speeds varying from the maximum to the slowest; for although most mercantile ships usually run at "full speed"—that is, nearly approaching the maximum—naval ships, cruising yachts, and some other vessels travel at varying speeds, and seldom for a long period at full speed. Such engines are by preference connected direct to the propeller shaft, but they may be again, as they have been in the past, geared to them by means of spur-wheels and pinions, in order that there may be no compulsion to drive the propeller at the same revolutions as the engines, or *vice versa*. The first screw steamers had engines similar to those in paddle-wheel ships, and moving at the same revolutions, and it is a fact that in the well-known trials of the s.s. "Rattler" *versus* the P.S. "Alecto," the engines in these ships were of the same size, design, and construction, but the screw of the "Rattler" revolved four times to one of the engines.*

Such engines as are suitable for the purposes of marine propulsion may be worked by means of *steam, or by gas* generated from coal or oil. Hitherto steam has been exclusively employed on sea-going craft, and oil vapour on

* *Vide* "The Screw Propeller," p. 202

small craft in home waters only. The success of the oil engine in such craft. and the fact that engines using only crude and non-dangerous oils can now be obtained, will no doubt lead to their use on large ships, and a more extended service now they are made reversible, so that the propellers can be reversed without wheel or other objectionable gearing, which when of small size do not develop defects so rapidly as in the case when larger power is put through them. The Diesel engine, which can be reversed by means of compressed air, and uses heavy oil, has been fitted to several ships, and is now being supplied to both the mercantile and naval marine in quite large sizes, by multiplying the number of cylinders. The suction gas plant, which on shore is used with fairly satisfactory results, may not be found so attractive on a service where the working day is twenty-four hours, and the working week seven days. It is, moreover, heavier than an oil engine installation.

The steam-worked machinery is more flexible than the oil and gas-worked, is less liable to derangement from shocks, and more easily governed in a sea way. But perhaps the most important feature to remember in making a comparison is the fact that steam is water vapour produced by heat generated by the combustion of anything that will burn, so that for the boiler of a steam engine fuel of sorts can be found in all parts of the world, whereas for the internal combustion engine only certain oils and certain coals can be used, and if they are not obtainable the engine is useless. There is, moreover, another feature that cannot be altogether overlooked, and that is, while the steam engine works with a back pressure of only 1 or 2 lbs. per square inch. the gas engine has over 15 lbs.

Marine steam engines are of two kinds, the one works by means of the elastic *pressure* of steam *during* expansion, the other by the *kinetic energy* of the steam developed *on* expansion ; in the one the action is static, in the other it is dynamic ; and just as in hydraulics, there is the common wheel with its buckets filled with water, whose weight causes it to turn ; likewise the ordinary ram which is forced outwards against a load, or as the reciprocating hydraulic engine when work is done by the *pressure* due to head, while the Pelton wheel is moved by a jet of water impinging on its vanes with *velocity* generated by *the fall* of the water, so there is the ordinary reciprocating expansive engine, and the turbine in which the steam acquires a high velocity by expanding into and through a nozzle, which directs its flow on to the blades of a rotor, where it gives up its energy by producing motion in it against resistance. The efficiency of a reciprocating engine is little affected by the velocity of flow of steam, be its piston speed high or low ; the revolution for maximum efficiency of the turbine must be such that the peripheral speed is half that of the flowing steam. This means that the revolutions of a simple turbine must be exceedingly high or the diameter of the rotor very large ; and for large power both. By methods adopted by Mr. Parsons and others, the rate of revolution has been brought down to reasonable limits for high-speed ships, while preserving a good efficiency ; but it is still too high for such ships as are engaged in cargo carrying if they are to have screws of decent efficiency and effectiveness. Mr. (now Sir Charles) Parsons has met this difficulty, therefore, by fitting pinions to the shafts of a pair of turbines, and geared them to a spur-wheel on the screw shafting.

The employment of a quick-revolution engine to drive the paddle shaft by means of a wheel gearing was practised by the Butterley Company so far back as 1823, but it was not followed till almost the end of the nineteenth century, when it was revived by one or two firms for river steamers, in order to use the quick-running triple-compound three-crank engine.

Wheel-gearing has found little or no favour in the eyes of the marine engineer since the geared screw engines of the middle of the nineteenth century were given up, and no doubt there will be much prejudice against it now, although it may be fairly urged that to-day the teeth are double-helical machine-cut, made of more suitable material, having no backlash, and the pinion driving instead of being driven (*v.* fig. 45). Experience condemned the system in the past, and it is the successful experience of Mr. Parsons which will revive it to-day.

With oil engines there seems to be, for practical reasons, a limit to the diameter of the cylinder (20 inches at present), and all increases of power is obtained by an increase in the number of cylinders. Moreover, since they are generally single-acting, and act at most once in two, and generally in four, strokes, their size for the power developed is large compared with that of the steam engine. The first oil engine was exhibited by its inventor at Cambridge 90 years ago. In 1817 Neepce proposed to use volatile oils

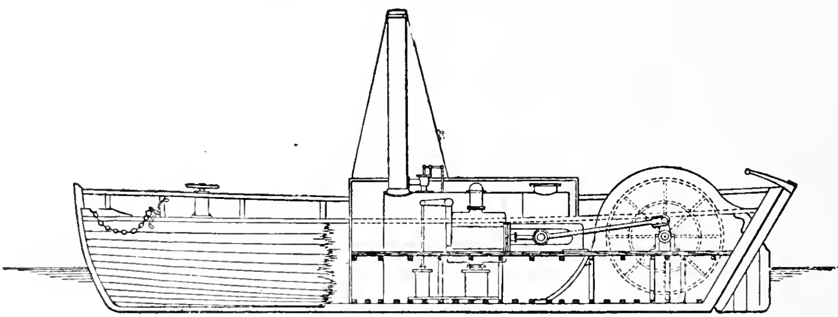


Fig. 3.—Paddle Steamer "Charlotte Dundas," 1802.

in his explosive engine for propelling ships, and an internal combustion engine was used so far back as 1825 to drive a screw propeller in a boat on the Thames. The system is even now only emerging from its infancy, and the room for developments is extensive. In the submarine ship there have been the greatest developments, as these engines are almost the only ones possible.

Steam used expansively.—The earliest marine engines were naturally near akin to those on shore, but it is very interesting and noteworthy that the one in the "Charlotte Dundas" of 1802 was a horizontal double-acting engine having a connecting-rod from the piston-rod end to the crank-pin, and, therefore, much in advance of and differing from entirely the engine of James Watt. The honour of designing the engine (see fig. 3) is due to William Symington, who patented the fitting of the connecting-rod, 1787, Pickard having taken out his patent for the crank and connecting-rod of the beam engine in 1780.

The "Comet," which was the first steamship in this country to earn money by conveying passengers and goods between Glasgow and Greenock

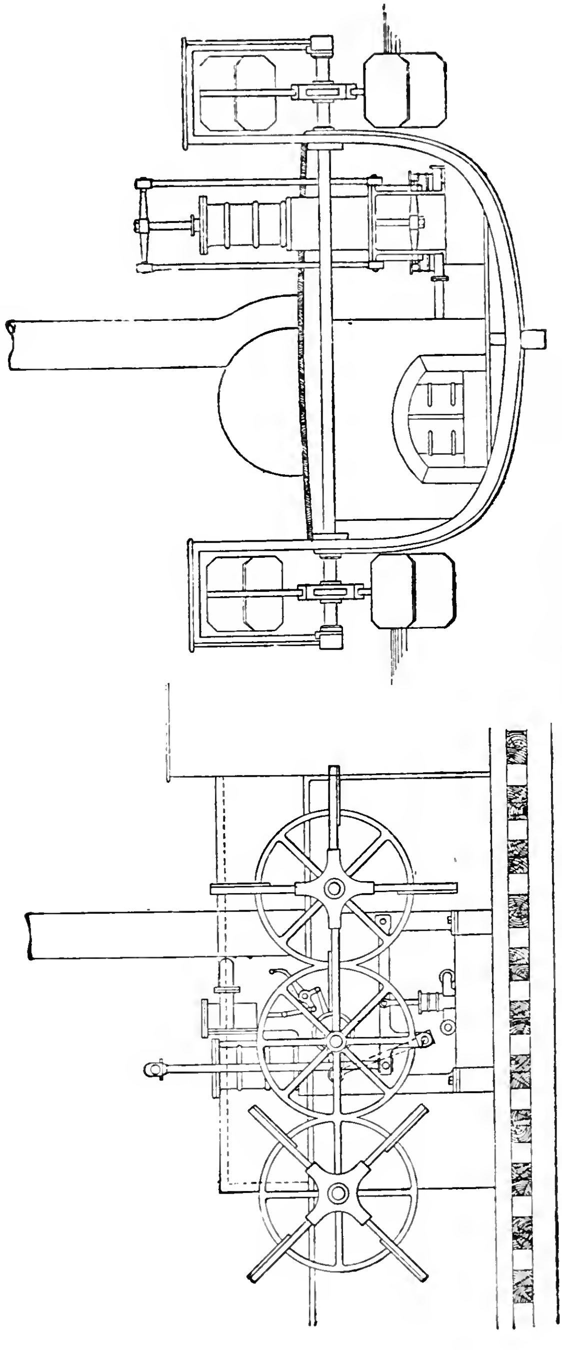


Fig. 4.—Machinery and Wheels of Paddle Steamer "Comet," 1812.

in 1812, had a modification of the beam engine of the type known later on as a *Side Lever* (*v.* fig. 4). These ships, as the ones that followed them, had engines using steam at or a little above the atmospheric pressure, and exhausting into a jet condenser with an air pump, etc. Later on, after the screw

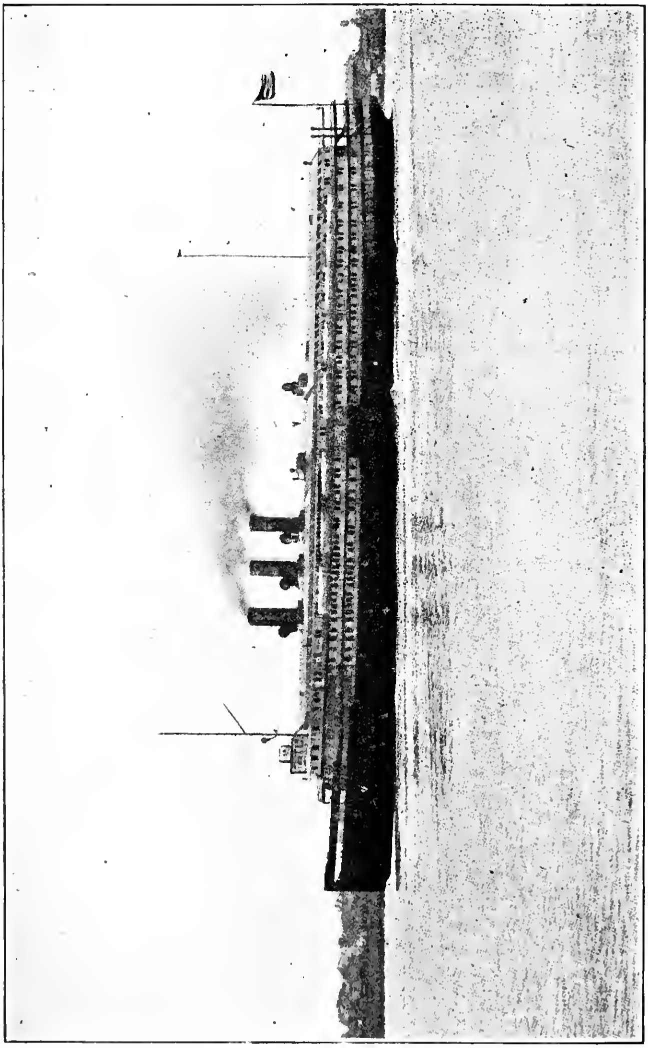


Fig. 5.—Lake P.S. "City of Detroit III.," Passenger Service. 6,370 tons displacement. 455″ × 55·3′ × 22″. 7,600 I.H.P. Speed, 18·27 knots.

propeller had been adopted, some small ships, and even a few large ones, in the Royal Navy, had non-condensing engines using steam at 50 to 65 lbs. pressure generated in cylindrical tubular boilers. Fig. 6 is a modern steamer as now built for service on the Clyde estuary, and doing the work that the

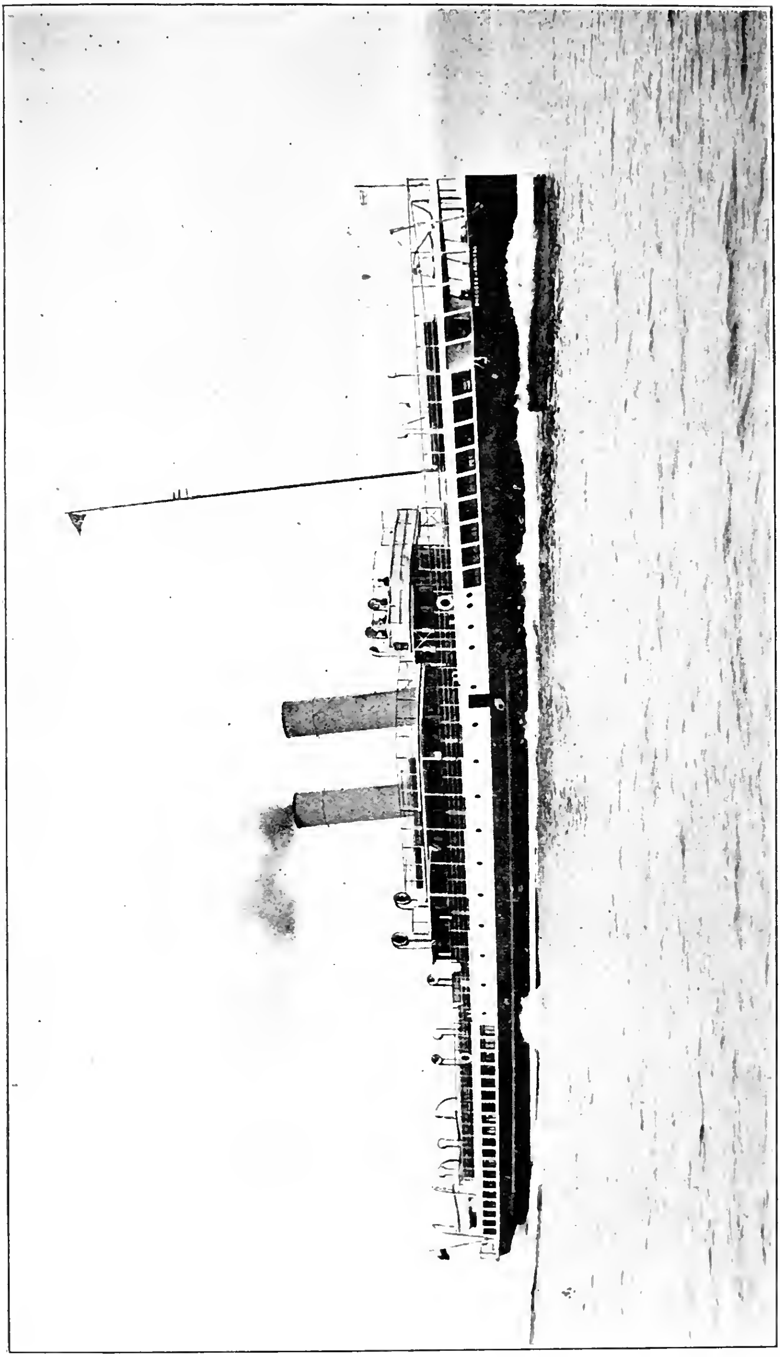


Fig. 6.—Turbine Steamer "Duchess of Argyll," 250 × 30 × 10·5. Speed, 21 knots.

"Comet" established; she is, however, driven by three screws and turbines instead of four paddles and a one-cylinder engine, and carries more passengers in a trip than the "Comet" in a couple of months.

In the early marine engines, as in those on land, the cycle was a simple one. Steam was admitted to one or more cylinders direct from the boiler during about 70 to 85 per cent. of the stroke, when working at full power; at or near the end of the stroke it was allowed to escape to the condenser; the cylinder on that side of the piston remained in open connection with the condenser during about 85 per cent. of the return stroke, and consequently its surface was exposed to the cooling action of the comparatively cold vapour remaining in it. The earliest screw ships in the Navy had their boiler safety valves loaded to 5 lbs. per square inch; by 1851 the load had been increased to 14 lbs., and two years later 20 lbs. was taken as the standard. In 1855 several special ships were fitted with non-condensing engines of considerable size supplied with steam from 50 to 65 lbs. pressure generated in cylindrical boilers of the so-called Scotch type, but the ordinary ships with condensing engines continued to work with steam generated in box boilers at 20 lbs. pressure. In 1861, the new ironclads had boilers with safety valves loaded to 25 lbs. per square inch, and four years after a further increase was made to 30 lbs., which was the usual or standard pressure till the box boiler ceased to be made.

The cut-off in the cylinders of these ships when running at full speed was never less than 60 per cent. of the stroke, so that the maximum rate of expansion was no more than 1·67 under these circumstances; but with the increase in pressure from 25 to 30 lbs. came the supply of expansion valves, whereby a much earlier cut-off could be obtained with a corresponding economy in fuel consumption. With such valves a rate of expansion of 4·0 could be obtained, so that the terminal pressure at which the steam exhausted to the condenser was under 10 lbs. absolute. In the mercantile marine about 1870 the boiler pressure was increased to 50 lbs. for expansive simple engines, and the cut-off at full speed was about 30 per cent. of the stroke, giving a rate of expansion of 3·33; a few ships with these engines had boilers loaded to 60 lbs., but in a general way 50 lbs. was the highest pressure with simple engines. The pressure at exhaust was then about 15 lbs. absolute at full speed.

The compound engine in which the steam, after doing duty in the first cylinder, exhausts into another, instead of delivering to the condenser, was the invention of Hornblower, a Cornish mining engineer, in 1771, improved by Wolff in 1804, and first used on shipboard by Randolph and Elder in the screw steamer "Brandon" in 1854 with a boiler pressure of only 22 lbs. This same enterprising firm supplied the first compound engines for H.M. Navy in 1863, and fitted them in H.M.S. "Constance." It is worthy of note that their designer was Professor Rankine, and they worked with a boiler pressure of 32·5 lbs. per square inch, and that the rate of expansion at full speed was about 5, and the referred mean pressure was 11·3 lbs. per square inch, which is not bad, considering that the theoretical would be not more than 22 lbs., or 51·5 per cent. efficiency.

It was not, however, till about 1870 that the compound engine was accepted as suitable and desirable for marine purposes, but after experience had proved its economy in consumption of steam, and that the expansive engine using high-pressure steam developed mechanical troubles from which it was free, it soon superseded all other engines, and was the accepted type until it was found that, with the increase of boiler pressure from 60 lbs. to 100 lbs., their gain in economy was not what engineers had reason to expect from such increases.

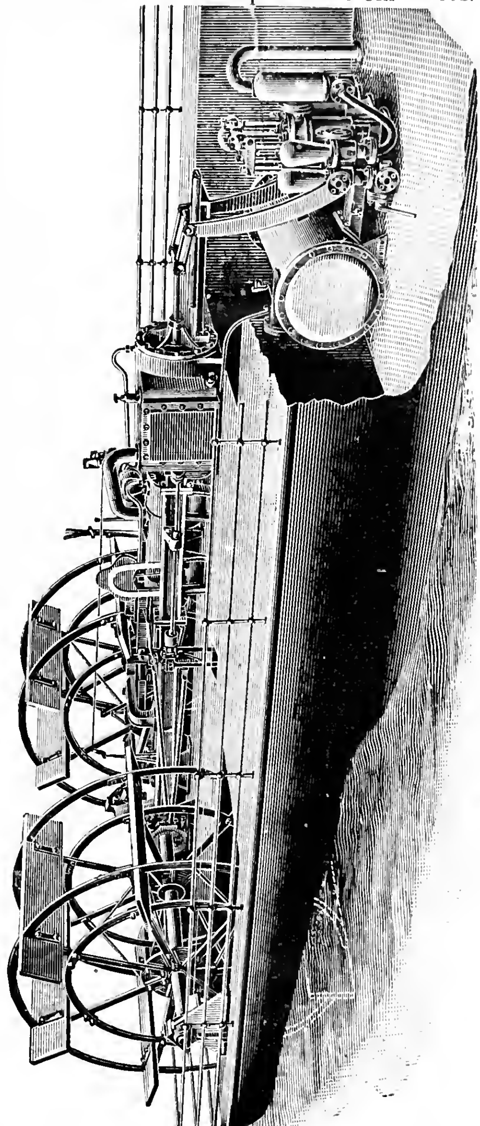


Fig. 7.—Engines and Paddle-wheels of a Stern Wheeler.

In 1881 the late Dr. Kirk, whose name is associated with that of John Elder, of the Randolph and Elder firm, and himself a most able and enterprising engineer, developed the idea of multiple cylinders, or, as we would now say, of expanding by stages; he built a three-stage engine, whereby the steam exhausted from the first engine into the second, and from the second to the third engine. Dr. Kirk had, however, in 1874 fitted an engine of this kind in s.s. "Propontis," with cylinders 23 to 41 inches, and 62 inches diameter by 42-inch stroke. Unfortunately, the water-tube boilers of this ship proved dangerous, and caused the whole installation to be doubted. Looking at the subject another way, it may be said that he introduced a third cylinder between the ordinary high-pressure and low-pressure cylinders, so as to avoid the big drop in pressure between the high and low. But the principle involved was one of stage, whereby there was a decrease in difference between the temperature during admission to, and that at emission of steam from each cylinder. There was, however, a mechanical gain as well, due to the decrease in initial and general loads on the pistons of the compound systems as compared with those obtaining in the expansive engines. The pressures of steam adopted by Dr. Kirk and by the author himself in the early triple-expansion engines was low compared with those now used; in fact, the pressure was little beyond that with which the compound engine was then working on shipboard. Since that time working pressures have gone on increasing till 180 lbs. was a common practice for triples in the mercantile marine, and on the introduction of the water-tube boiler into the Navy, steam of 250 lbs. pressure was and is used by the triple-compound engine. In the mercantile marine now steam up to 225 lbs. is generated in the cylindrical or tank boiler for use by quadruple expansion engines with advantage; these engines were introduced by two or three of our leading manufacturers in 1885. The rate of expansion had gone on increasing, till now, with quadruple-expansion engines it is common practice to work at 16, while with the triple engine working with a boiler pressure of 180 lbs. the rate of expansion will be 12 to 13 in the mercantile marine, and about 11 to 12 in H.M. Navy at full power.

Until quite recently all attempts at economy in consumption of fuel were in the direction of increase of pressure and increasing the number of cylinders. The upper end of the steam expansion diagram was the field in which further gains were looked for, and the dictum of the old chief engineers that there was no economic gain in working with a vacuum over 24 inches was accepted without a close enough examination of the foundations for such a statement, hence few, if any, engineers turned their attention to the possibilities of the gain to be got at the lower end (*v.* fig. 66). The adoption of the turbine for marine propulsion by Mr. Parsons opened his eyes to this by the great increase in efficiency of his machines when working with the high vacuum so cheaply obtained with an unlimited supply of cooling water and a really good air pump on shipboard; and further, the general use of feed heaters on shipboard has quite removed the bugbear of cold-feed water for the boilers. Before 1865, with the old common jet condenser, the vacuum was seldom more than 24 inches, whereas with the surface condenser, which soon became the common practice after that date, 28 inches was not uncommon, and 27 inches easily maintained; the Admiralty requirement was that the vacuum on trial should be within 3 inches of the barometer.

In passing, however, it may be mentioned that the pressure in the condenser is in no way subservient to that of the atmosphere, although those registered by the common vacuum gauges, of course, are, for they really only exhibit the difference between the pressure of the air and that in the condenser. To-day gauges can be obtained which indicate the exact pressure in a condenser, without regard to the atmospheric pressure, and should be always used.

It was not an unknown thing forty years ago to maintain, in a surface condenser, a vacuum of 29 inches with high barometer (30·5 inches), but it was only with one of exceptionally good design, and it, the pumps, and everything in the best of order that so good a vacuum was got. To-day, with the better design of both condenser and pumps, and the relay system or other means for obtaining high efficiency of air pumps, 29 inches can be maintained with comparative ease, consequently a rate of steam expansion that was useless formerly may now be worked with advantage and with considerable gains both in power and economy, especially with turbines.

The reciprocating engine has its limitations, one of which is the size and nature of the valves for admitting and releasing the steam. It is practically impossible to provide means in this direction to ensure high efficiency to the low-pressure cylinder of a compound system; wire drawing and clearance losses are great, and the difference in pressure between the cylinder and the condenser necessarily great compared with those of a turbine. Hence the last addition to a marine engine, whereby steam efficiency is considerably enhanced, is the low-pressure turbine taking its steam from the exhaust pipe of the low-pressure cylinder, and expanding it from about 10 lbs. to 1 lb. absolute, or even less. It is claimed by Messrs. Denny that the increase of power and an economy of fuel has been found as high as 15 per cent.

The compound turbine so much in use to-day in H.M. Navy and in express steamships (*v.* fig. 6) of high speed is capable of a rate of expansion far beyond that of a reciprocating engine; in fact, there is no practical limit to it. But in the early stages of the compound turbine the thermal efficiency is not so high as is the triple compound engine; it follows then that the maximum efficiency will be obtained by the combined arrangement of a reciprocator with a low-pressure turbine, taking the steam from the low-pressure cylinder at something like 15 lbs. pressure absolute, which is about the terminal pressure of such an engine working at 200 lbs. boiler pressure.

Propellers.—The "Charlotte Dundas" (fig. 3), the first ship to be propelled by steam in a practical way, had one paddle-wheel at the stern. The "stern wheeler," as she is now called, remains as the surviving representative of the paddle ship in the construction of steam craft for river purposes, especially those working in the tropics, where she is the favourite, and appears likely to continue as such. The first steamer in America, the "Claremont," of 1807, and the "Comet" in this country in 1811, as the first British steamer to be put to commercial purposes, had a pair of side wheels—that is, a wheel on each side with an axle or shaft common to the two. It should, however, be noted that the "Comet" had two pair of wheels (*v.* fig. 4) when first tried, a system followed by Sir Edward Reed in 1874 for special reasons when he designed the swing saloon steamship "Bessemer." The pair of side wheels continued to be the practice for general purpose down to recent times, and is still followed when such ships are built for the special services already alluded to. The last of the ocean-going paddle ships was the "Scotia," 6,871 tons displacement, 362·5 feet long, having engines of 4,950 I.H.P., driving a pair of wheels 40 feet diameter, weighing 156 tons. The largest modern American paddle steamship is the "Priscilla," 424 feet long and 5,200 tons displacement; she has engines developing 9,345 I.H.P., which drive her at a speed of over 19 knots per hour. Fig. 5 shows the paddle steamer as now constructed in America for lake and river service to convey passengers and parcels expeditiously. The largest modern British paddle steamship (*v.* fig. 8) is the "Empress Queen," 360 feet long, 2,900 tons displacement, attaining a speed on service of 21·7 knots with 11,440 I.H.P. The fastest British paddle steamship is "La Marguerite," 330 feet long, 1,868 tons displacement, and having a speed of 22·3 knots.

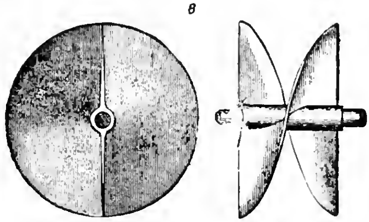
Screw propellers, brought into practical use by Bennet, Woodcroft, Francis P. Smith, and John Ericsson, in 1836 (*v.* fig. 9), are now the most important instruments of propulsion, and, although much time, genius, and money have been expended in exploiting screws of the most varied forms, engineers have to-day settled down to the almost universal use of the screw, such as used by Smith and his friends in 1840 on the s.s. "Archimedes"

Fig. 8.—P.S. "Empress Queen," 360′ × 42′ 3″ × 17′ 0″. 3,015 tons displacement, 11,000 I.H.P. Cylinders, 92-68-92 ins. diam. and 84 ins. stroke.

—that is, in the essentials—viz., making it a portion of a true helix and of moderate diameter and acting surface. So far as shape is concerned, the practice now is very like that of Robert Griffiths of 1865—that is, the blade is narrow at the tips compared with the breadth at middle, and the boss spherical of considerable diameter. The prevailing form for express and naval ships is, of course, a modification of the Griffiths, inasmuch as their tips are somewhat fuller and rounded, so that the blade is the shape of the longitudinal section of a domestic hen's egg; and in some cases where the diameter is small for the power, the blade is even circular. In fact, all of them are such as would be formed by taking the Griffiths, and reduce the diameter by rounding the tips until the maximum breadth is nearer the top.

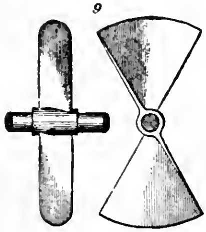
The number of screws has increased from the one situated in a gap in the deadwood of the ship to the four of the modern high-speed warship driven

8



Francis Pettit Smith, 1836.

9



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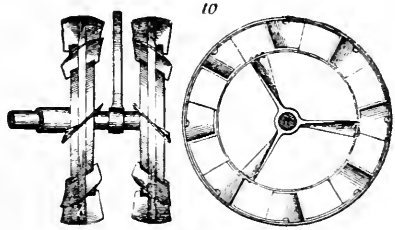


Francis Pettit Smith, 1838. John Ericsson, 1836.

Fig. 9.

by turbines. The twin screw, an obvious development, especially if required to compete with the paddle-wheel in shallow waters, was first introduced by the Rennies in 1854 for service on the Nile; since then they have replaced the single screw in all express and ocean-going high-speed steamers. With such twin screws, not only is total immersion attained and the "feed" to them unobstructed, but the safety of the ship considerably insured, inasmuch as the liability to total breakdown is reduced by a-half, and a means of steering is provided, should the rudder or its gear be disabled. Two screws, one at the bow and another at the stern with a line of shafting common to them have been used frequently for tugs and ferry boats with advantage, and four screws similarly arranged have been used for the same services.

Three screws, one in the deadwood and one on each side, somewhat ahead of the middle one, have been used in France, Germany, and Italy, and the United States, but the system did not find favour in this country till the turbine was used as the prime mover. The development is a natura one, where large power is required in a comparatively shallow vessel, and also with the view of keeping the engine within moderate dimensions. Italy was the first to give the system a practical trial in 1886 on the cruiser "Tripoli"; France followed on a large scale with the "Dupuy de Lome" in 1890. In 1892 the United States of America adopted the system for the "Columbia," of 7,375 tons displacement, 22·8 knots speed. In the same year the German Government fitted the cruiser "Kaiserin Augusta," of 6,330 tons, and 22·5 knots, with three screws, while in 1896 Russia adopted the system for the "Rossia," a large cruiser of 12,130 tons and 14,500 I.H.P. In our own Navy, the "Amethyst," cruiser of 3,000 tons, having Parsons turbines operating on three screws, was the first attempt, if the experiment with H.M.S. "Meteor" in 1855 is excepted. To-day the rule in the British Navy is to fit four screws* to all but the very small ships; the newest cruisers of 75,000 H.P. are so fitted, as are the new class of "Destroyers." In the mercantile marine our largest and fastest mail steamers have four screws; the smaller ones, three when driven by turbines or combined reciprocators and turbines. The newest and largest ship, s.s. "Olympic," has thus three screws (*v.* figs. 43 and 44). With reciprocators only the twin screw is now almost the universal rule for all express steamers, and for cargo steamers of the largest size.

Multiple Screws.—In some ships of special design, or for special service, a larger number of screws have been fitted, and their arrangement varied; for example, the saucer-shaped Russian warships "Popoffskas" had six screws, and the Russian ice-breaker, "Ermack," has three screws at the stern and one at the bow.

Mr. Parsons has also in some few cases fitted more than one screw on each shaft, and more recently the German Naval Authorities have tried, on the cruiser "Lubeck," eight screws, two on each of her four shafts; the results of this experiment are by no means satisfactory, being much inferior to those of sister ships having twin screws driven by reciprocating engines.

Table I. contains examples of naval ships' engines typical of the periods since the introduction of the screw propeller, and shows the progress made in the use of steam expansively.

Progress in the use of steam expansively in the *mercantile marine* during the past 50 years, as shown by the typical examples in Table II.

* Quite recently, however, there is a tendency in both the British and Foreign Navies to revert to the twin-screws for the smaller class of ship, and even in second-class cruisers, with a complete turbine for each propeller, and since the introduction of gearing the tendency is to twin-screws for all kinds of ships.

TABLE I.—PROGRESS OF NAVAL ENGINEERING (SCREW PROPULSION).

Date.	Name of Ship.	Cylinders.			Indicated Horse-Power.	Revolutions.	Boiler Pressure.	Approximate Rate Expansion.	Actual Mean Pressure.	Theoretical Mean Pressure.	Engine Builder.
			In. diam.	In. stroke.			Lbs.		Lbs.	Lbs.	
1844	Rattler, . .	4	40	× 48	428	26·0	5·0	1·25	13·4	...	Maudslay & Field.
1850	Simoom, . .	4	43·4	× 36	540	52·5	10·0	1·33	11·5	...	Bolton & Watt.
1850	Blenheim, . .	4	52	× 36	938	43·0	10·0	1·25	14·1	...	Seaward & Capel.
1855	Brunswick, . .	2	64	× 36	1,438	53·5	20·0	1·33	23·0	32·0	Miller & Ravenhill.
1855	Malacca, . .	2	= 28·4	× 30	692	87·5	60·0	1·60	41·2	54·0	J. Penn & Son.
1857	Urgent, . .	2	64	× 36	1,020	56·3	20·0	1·88	15·3	26·0	...
1860	Mersey, . .	2	92	× 48	4,045	55·3	20·0	1·33	22·7	32·0	Maudslay.
1861	Octavia, . .	3	66	× 42	2,265	69·4	20·0	2·00	15·0	27·0	Maudslay.
1862	Galatea, . .	2	= 82	× 44	3,517	57·5	22·5	1·43	19·0	...	J. Penn & Son.
1863	Constance, . .	6 Comp.	78-60-78	× 39	2,301	54·0	32·0	5·00	11·3	22·0	Randolph & Elder.
1865	Agincourt, . .	2	101	× 54	6,867	61·5	25·0	1·50	25·6	34·0	Maudslay.
1865	Northumberland,	2	= 109	× 52	6,543	58	25·0	1·56	22·9	32·8	J Penn & Son.
1865	Pallas, . .	4 Comp.	51 & 99½	× 39	3,630	70·4	32·0	4·50	16·9	24·0	Humphrys.
1866	Bellerophon, .	2	= 104	× 48	6,000	73·0	30·0	1·82	20·0	37·0	J. Penn & Son.
1867	Lord Warden, .	3	91	× 54	6,706	63·3	31·3	2·00	19·2	36·0	Maudslay.
1870	Inconstant, . .	2	= 104	× 48	7,361	74·5	30·0	1·66	23·9	38·0	J. Penn & Son.
1870	Monarch, . .	2	120	× 54	7,842	63·0	31·6	1·84	20·2	37·0	Humphrys.
1870	Captain, . .	4 Twin S.	80	× 39	5,990	74·3	30·0	1·84	20·0	37·0	Laird.
1871	Tenedos, . .	2 Comp.	56·7 & 90	× 30	2,018	99·7	50·0	6·0	21·9	29·0	J. Elder.
1871	Coquette, . .	2 ,,	31 & 48	× 18	410	126	60	7·0	20·7	31·0	Jas. Watt.
1880	Battleship, . .	Twin	60 & 104	× 42	6,624	82·1	65·0	6·6	22·3	33·0	...
1885	,, . .	,,	77-55-77	× 48	10,184	80·1	90·0	6·5	25·8	43·5	...
1886	Cruiser, . .	,, 4	38 & 64	× 39	6,151	120·0	110·0	5·0	40·4	62·5	...
1890	Battleship, . .	,, Triple	43-62-96	× 51	12,465	95	135	7·0	34·8	61·5	...
1890	Cruiser, . .	,, ,,	33-40-74	× 39	9,435	140·0	155	8·5	40·0	61·5	...
1900	,, . .	,, ,,	34-55-64-64	× 48	16,961	116·6	300	12·0	46·7	74·0	...
1905	Battleship, . .	,,	38-60-67-67	× 48	18,500	120·7	190	9·0	45·0	71·0	...
1905	Cruiser, . .	,,	43½-69-77-77	× 42	23,500	135·4	190	10·5	45·0	64·5	...
1905	Scout, . . .	,, 2	31½-50½-57-57	× 30	14,700	211·6	227	12·0	44·9	68·0	...

TABLE II.—Progress of Engineering in the Mercantile Marine.

Date	Name of Ship.	Cylinders.			Indicated Horse-Power.	Revolutions.	Boiler Pressure.	Approximate Rate Expansion.	Actual Mean Pressure.	Theoretical Mean Pressure.	Coal per I.H.P. Hour.	Service.
			In. diam.	In stroke			Lbs.		Lbs.	Lbs.	Lbs.	
1850	Atln., .	2	65	× 42	1,320	45·0	24	1·67	18·0	32·0	3·30	North Atlantic
1850	Ocnml.,	2	90	× 42	2,500	46·0	20	1·33	20·0	32·0	3·04	Do.
1860	Pcfm., .	4	48 & 93	× 60	1,350	21·0	30	5·50	15·6	21·0	2·07	Pacific S.
1870	Sprbr., .	2	37·5	× 36	670	65·0	48	4·80	25·6	31·5	2·7	North Sea.
1870	Ydo., .	2	33½ & 67	× 36	707	57·0	65	6·25	19·2	32·7	2·5	Baltic.
1875	Slir., .	2	52 & 96	× 48	2,180	60·0	63	5·50	20·8	37·0	2·3	North Atlantic.
1875	Brtnc., .	4	48 & 83	× 60	5,400	56	70	5·0	28·7	42·0	2·3	Do.
1880	Nrthn., .	2	26 & 56	× 60	1,241	57·5	115	7·7	30·0	50·0	2·1	General Trade.
1880	Arzna., .	3	90-62-90	× 66	6,300	..	90	7·0	27·0	42·0	1·95	North Atlantic.
1880	Srons., .	2	42-84	× 60	2,347	57	90	7·0	24·5	42·0	1·95	General Trade.
1885	Umbra.,	3	105-71-105	× 72	14,300	..	110	7·0	33·6	50·0	1·85	North Atlantic.
1885	Mrtlo., .	3	31-50-82	× 57	2,408	58·2	150	12·0	27·2	45·0	1·67	Do.
1885	Ornco., .	3	42-62-96	× 66	5,556	64·0	145	8·5	36·0	58·0	1·65	West Indies.
1885	Cntyk., .	4	20-28-40-57	× 42	682	48·5	160	13·5	26·3	43·7	1·60	General Trade.
1890	Mjstc., .	6	43-68-110	× 60	17,000	78·0	180	10·0	37·8	62·0	..	North Atlantic.
1890	Plo.,. .	4	12·7-18·5-27-39	× 24	495	98·0	200	15·0	34·9	50·6	1·45	North Sea.
1895	Knsntn.,	8	25·5-36·5-53-75·5	× 54	8,300	87·0	200	19·5	40·0	58·0	..	North Atlantic
1895	Dido., .	3	24·7-40-70	× 48	2,430	71	200	13·5	36·9	56·0	1·50	Indian Trade.
1900	Ocnc., .	8	47·5-79-98-98	× 72	28,000	79	192	13·0	35·8	55·0	1·40	North Atlantic
1909	Kwlm., .	16	37-49-75-112	× 71	40,000	80	225	15·0	35·0	58·0	..	Do.
1914	Brinc., .	8	54-84-97-97	× 75	53,000	77-170	215	Recipros.	and Turb	ine	1·32	Do.

CHAPTER II.

RESISTANCE OF SHIPS AND INDICATED HORSE-POWER NECESSARY FOR SPEED.

Although, strictly speaking, it is not the province of the engineer to determine the power necessary to drive a ship at a certain speed, but rather that of the naval architect, still it is a point of great importance to the engineer, and one with the investigations of which he should be fully acquainted. Circumstances sometimes require, indeed, that the engineer shall name the power, as the naval architect may submit that, inasmuch as he is unaware of the efficiency of the particular engine to be supplied, he cannot say what *indicated* horse-power will be necessary, but only what *effective* horse-power. Moreover, the subject is one possessing great interest at all times, and sometimes of the utmost importance to the engineer, as the deficiency of speed obtained at the measured mile from that anticipated may be attributed to the inefficiency of the engine and propeller. This charge may be, and often has been, proved to be true; but, on the other hand, it may be without foundation, the blame really belonging to the designer, who has given the ship lines unsuited to the speed.

Value of Trial Trips.—Trial trips are now conducted, both in the mercantile marine and the Royal Navy, with more care and interest than obtained formerly; and it is not sufficient to prove at the measured mile merely that the ship has done the speed expected, or that the engines have developed the power for which they were designed. Both engineers and naval architects desire to determine whether the speed has been obtained with the minimum of power, and the engineer can satisfy himself on a most important point—viz., the efficiency of the propeller, and, to some extent, the efficiency of the machinery, while the owner, if it be a private ship, is enabled to judge whether he is paying for what he calls "big horses" or "little horses." Another point (and one most important to the owner) which, to some extent, is determined on a trial trip is—at what expenditure of fuel a ton of displacement is carried over a mile. It is not an unknown thing to find that the engine which burns least fuel per I.H.P., does not compare so favourably with others when measured by this latter standard. The apparent contradiction here is not very difficult to understand when fully looked into; it may be, perhaps, best comprehended by taking extreme cases. Suppose the blades of the screw are set so as to have *no* pitch; the engine will work, develop a certain power necessary to overcome its own resistance and that of the screw, but it will not drive the ship an inch; the coal consumption per I.H.P. will probably be somewhat heavier than that of the same engine when working with half its load, but still may be light. Now place the blades fore and aft, so that the pitch is infinity, and although there may be now a large development of power, there will be no appreciable speed—theoretically, none at all. In both these extreme cases the consumption per I.H.P. may be very satisfactory, but the satisfaction would not be experienced by the owner. It is manifest, then, that between these two extreme limits of pitch there is some value and one position of blade which will give the best result, so far as economy of fuel for load propelled is concerned. Not only is the pitch of propeller an important function in all calculations relating to the speed of ships, but the diameter has a very important bearing also on the subject, and more than was generally thought previous to the remarkable trials of H.M.S. "Iris."

The Resistance of a Ship passing through water is not easily determined beforehand, as it may vary from more than one cause, and in a way often unanticipated, as has been seen during the trials of the very fast torpedo boats and destroyers. The investigations of the late Dr. Froude on this subject have shown that the older theories were sometimes erroneous, and the old-established formulæ unreliable; and perhaps the best source of information on the intricacies of this somewhat complex subject is to be found in the many able papers read by him, and others since his day, before the Institution of Naval Architects and other learned societies.

When the screw or paddle first commences to revolve, the ship makes no headway, and it is only after some seconds have elapsed that motion is observable. The engine power has, during that period, been employed in overcoming the resistance to motion which all heavy bodies possess, and which is called the *vis inertiæ*. When the engine is stopped at the end of the voyage, the ship will continue to move, and come gradually to rest, unless otherwise retarded by the reversal of the engine or by check ropes. The ship is then said to have "way on her," a phrase which, in scientific language, means that she possesses stored-up energy, called *momentum*, which is given out, when the engine stops, in overcoming the resistance of the water to the passage of the ship through it. This energy was stored up at starting of overcoming the inertia, and remains stored until there is any retardation in velocity. In this way the weight of the ship helps to preserve a uniformity of motion, as that of a flywheel does to an engine, and, therefore, it is important that tug-boats should have weight as well as power, to prevent towing in the jerky fashion so often observable. When the *vis inertia* has been overcome, the power of the engine is directed to overcoming the resistance of the water, and wind if there be any, and in accelerating the velocity of the ship; as the speed increases, the resistance much more increases, until the surplus power available for acceleration becomes *nil*, and the whole engine power is absorbed in overcoming the internal resistances, or those belonging to the engine itself and the propeller, and the external, or that of the ship.

The Resistance of the Water is Twofold.—First, the ship in moving forward has *to displace a certain mass of water of the same weight as itself*, and the water has to fill in the void which would otherwise be left by the ship. The work done here is measurable by the *amount* of water, and since it is equal to the displacement of the ship, displacement becomes a factor in the calculations of resistance. But to effect this displacing and replacing of water with the least amount of energy, it is necessary to do it gently—to set the particles of water gradually in motion at the bow, and let them come gradually to rest at the stern. If it is not done gently, and the water is rudely separated, a wave is formed on either side, showing that energy has been spent in raising the water of this wave above its normal level. Although every ship, however well designed to suit the intended speed, causes these waves of displacement, it is the object of the naval architect to reduce their magnitude as much as possible.

The chief cause of resistance to the passage of a ship through the water is, however, the friction between the surface of the immersed portion and the water. Resistance from this cause is generally spoken of as *skin resistance*, and is in well-formed ships much greater than the resistance due to other causes. However fine a ship may be, there is, of necessity, a certain area of skin exposed to the water, and though the displacement be very small indeed, and the section transverse to the direction of motion reduced to a minimum, it is found that a considerable amount of power is required to propel the ship through the water, and that, roughly, the power is proportional to the *wetted surface* at the same speeds. It is from this cause that the older rules for speed, involving only displacement, or area of midship section, together with speed as variables, are found to be so misleading.

Residual Resistance is the term generally used to express the sum of all other resistances to be overcome in propelling a ship through the water, and includes that due to wave-making, eddy-making, etc. In tank experiments this is differentiated from skin resistance, and ascertained with accuracy, but it may be calculated with a very fair approximation to it by a formula arrived at by Mr. D. W. Taylor, U.S.A., and published by him there. It is as follows, viz. :—

$$\text{Residuary resistance in lbs.} = \frac{12{\cdot}5\, b\, D\, V^4}{L^2},$$

where b is the block coefficient;
D is the displacement in tons;
V is the speed in knots;
L is the length on the water-line in feet.

The formula is applicable only to speeds for which $\frac{V^2}{L}$ is less than 1·2. As will be seen, it conforms to the law of comparison, and, compared with other formulæ, gives extremely good results for all classes and sizes of ships. It has been applied to a large number of different ships, and the general results of the application are satisfactory.

Most of the examples taken for comparison have been warships, where it was found in applying the formula that a slight modification gave better results. This modification consists in taking the length as the extreme length of immersed vessel in place of that on the water-line, and in calculating the block coefficient on the extreme length instead of that between perpendiculars, as usually taken. For merchant ships the two lengths are generally the same, but for warships the difference in the two is appreciable. Mr. A. W. Johns found that for vessels in which the block coefficient varies from ·6 to ·65, the calculated results are generally correct for a speed such that $\frac{V^2}{L}$ is about ·85. Below this speed, the results are generally about

$7\frac{1}{2}$ per cent. larger, whilst above this speed the results are somewhat smaller than the experimental results.

For vessels in which the block coefficient varies from ·5 to ·55, the calculated results are generally correct for a speed given by $\frac{V^2}{L}$ equal to 1. Above this speed, the results are slightly smaller, whilst below they are generally about 10 per cent. greater than the experimental results.

The resistance of the ordinary ship roughly varies as the square of the speed, so long as the highest speed does not exceed that for which the ship is suited, and so long as the wave formation is not so great as to cause considerable variation in the *trim* of the ship. When such conditions prevail, as they do largely in modern very high-speed ships of small size as are Destroyers and Scouts, the variation in resistance does not follow so simple a rule, and, moreover, the fluctuation is apparently capricious, as may be seen by examining the figures given by Sir William White for destroyers (*q.v.*). Assuming, however, as we may in the case of ordinary ships, that R varies as S^2, then to complete the expression, which will give a definite value to the resistance of a given ship, it was necessary to multiply the product of the above two variables by a quantity found from practice; and if the law were absolutely correct, this quantity should have a fixed value, whatever the size and form of the ship, and would be a "*constant*" multiplier for all cases in choosing values for C and K. Actual values for them can be found in the tables of performances of ships on trial trips.

Coefficient of Fineness.—To determine the form of a ship, as to whether it is "fine," "fairly fine," or "bluff," it is usual to compare the displacement in cubic feet with the capacity of a box of the same length and breadth, and of depth equal to the draught of water; the coefficient by which the capacity of such a box must be multiplied to give the displacement being called the *coefficient of fineness.* Thus

$$\text{Coefficient of fineness} = \frac{D \times 35}{L \times B \times W}.$$

D being the displacement in tons of 35 cubic feet of sea-water to the ton; L the length between perpendiculars in feet; B the extreme breadth of beam in feet; and W the mean draught of water in feet, less the depth of the keel. Strictly speaking, the length should be measured from the stem to aft part of body-post on the water-line, instead of to aft part of rudder-post; but as this dimension is not easy to ascertain without referring to the plans, and the calculation is made for the sake of comparison, rather than as an accurate computation, no inconvenience will arise from this, so long as all the ships under comparison are measured in the same way.

It will be easily seen that the above coefficient only expresses a relation between the cubic contents of the immersed portion of the ship and a box of the same dimension, and gives no certain clue to the fineness of the *water-lines,* which is really what is wanted for consideration in dealing with the question of power for speed.

Two ships may have the same dimensions and the same displacement, and, consequently, the same coefficient of fineness, and yet one may have bluff lines and the other fine—the difference arising from the latter having a flat floor, and the former a high rise of floor. To take an extreme case, the fine ship might have a rectangular midship section, and the bluff one a triangular one; and if the "coefficient of fineness" was 0·5, the bluff ship would have rectangular water-planes, while those of the fine ship would be two triangles base to base.

Now, if a coefficient be obtained by comparing the displacement with the volume of a prism, whose base is the midship section, and height the length of the ship, it will indicate the general fineness of water-lines, and form a guide in the choice of the constants for speed calculations.

$$\text{The Prismatic or Coefficient of water-lines} = \frac{D \times 35}{\text{area of immersed mid-section} \times L}$$

The Skin Resistance is, in all classes of ships, the most serious, although in destroyers at full speed (say 30 knots) it only amounts to 45 per cent. of the total, as compared with 80 per cent. at 12 knots. With cruisers it is as much as 80 per cent. at 20 knots, and more than 70 per cent. at the full speed of 23 knots, while at 12 knots it is 90 per cent.

From experiments made by Dr. Froude with varnished surfaces, such as given by the modern spirit mixed anti-fouling compositions applied to ship's bottoms to discover n, the index of V, the following was deduced :—

$$\text{Resistance} = j \times A \times V^n.$$

j is a factor which varies with the length of the ship.
A the area exposed to water rubbing.
V the velocity in feet per second.
n had a value 2 close to the bow, 1·85 at 20 feet from it, and 1·83 at 50 feet. The average value throughout may be taken at 1·83.

At a speed of 10 feet per second the mean resistance was found to amount to 0·25 pound per square foot.

Taking as an example an express steamer, 400 feet long, at a speed of 20 knots, what will be the resistance per 100 square feet of wetted skin? Here j = ·00886, and V = 34.

$$R = 0{\cdot}00886 \times 100 \times 34^{1{\cdot}83} = 563 \text{ lbs.}$$

The power required to draw this surface through the water at the velocity will be

$$\text{Power} = 563 \times 34 \times 60 = 1{,}148{,}520 \text{ foot-lbs., or } 34{\cdot}8 \text{ H.P.}$$

This, of course, is the net horse-power, and not that developed by the engine driving the propeller. If, however, the efficiency of machinery, propeller, etc., were, say, 0·7, then

$$\text{The gross I.H.P.} = 34{\cdot}8 \div 0{\cdot}7 = 49{\cdot}7.$$

The Wetted Skin of a Ship should be accurately measured, but this is a somewhat long and troublesome business, and, moreover, necessitates having the full lines of the ship, which, as a rule, are not available in the initial stages of design, and certainly not accessible to the engineer as a rule. There are, however, methods of obtaining the area with sufficient accuracy for the purpose of estimating the indicated horse-power necessary to drive a ship at a required speed, and certainly is accuracy sufficiently close if the allowances have been obtained from practice with wetted skins calculated by the same methods.

Kirk's Analysis is a system introduced by the late Dr. Alexander Kirk, whereby the qualities of ships can be compared and incidentally the fitness of a proposed ship for the speed required. By the means he provided the wetted skin is found, and for ships with a high rise of floor, as were the rule at the time the area so calculated was within 3 per cent. of the actual, and often almost identical with it. With modern ships the error is often as much as 5 per cent., and even with ships having deep bilge keels and large fins for the side screws the error is 3 per cent.

Mumford's method of calculating wetted skin gives fairly accurate results with ships of normal form and proportions, but with shallow draft ships and flat bottoms the actual surface is somewhat in excess of that given by his rule, which is as follows :—

$$\text{Wetted skin} = \text{L}\ (1{\cdot}7\, d + b\, \text{B}).$$

L is the length between perpendiculars, d is the depth of immersed midship section, B is the greatest beam, and b is the block coefficient of displacement.

Seaton's Modification of Mumford's method is as follows :—

$$\text{Wetted skin} = (c \times d \times \text{L}) + \frac{\text{D} \times 35}{d}.$$

Where c is a coefficient $= 2 \times$ area immersed mid-section $\div \text{B} \times d$. For flat bottom shallow-draft ships c is 2·0, while for ships with a high rise of floor, as usual in yachts and fast-sailing ships, c is 1·6. For ordinary ships with a draught of water not less than one-quarter the beam c is 1·8.*

D is the displacement in tons, and d the mean moulded draft of water.

Seaton's Method for sea-going ships, whose draft of water is more than one-quarter the beam permits of a ready computation of wetted skin with the same information; here L is the length, B is the beam, and d the moulded draft.

(1) $\text{K} = \text{L} \div (0{\cdot}55\, \text{B} + d)$.
(2) $\text{F} = 42 \sqrt[4]{\text{K}}$.
(3) Wetted skin $= \text{F} \times \text{D}^{\frac{2}{3}}$—that is $= 42 \sqrt[4]{\text{K}} \times \text{D}^{\frac{2}{3}}$.

The following are the values of F for variations in K :—

When K is 4, the value of F is	59·4	When K is 8, the value of F is	70·6
,, 5, ,, ,,	62·9	,, 9, ,, ,,	72·8
,, 6, ,, ,,	65·7	,, 10, ,, ,,	74·7
,, 7, ,, ,,	68·3	,, 11, ,, ,,	76·4

The form suitable for a required speed is gauged by the coefficient of fineness of water lines, called usually the prismatic coefficient, as already stated; it also may be obtained from the block coefficient by dividing it by the coefficient of mid-ship section, when knowing the area of midship section.

$$\text{Prismatic coefficient} = \frac{\text{Displacement} \times 35}{\text{Area mid-section} \times \text{length}}.$$

This may be called the *criterion of form.*

* To ships with multiple screws c is increased by $3\frac{1}{2}$ per cent., so that the multiplier is 2·07 instead of 2·0.

The author has devised a rule for guidance in designing a ship, as also to provide a means whereby engineers may avoid trying to do impossible, or, at least, non-economic things in the way of forcing a ship beyond her capacity for speed.

Seaton's Rule for Limitation of Speed is as follows :—

$$\text{Suitable prism coefficient } F = 0{\cdot}4 \sqrt[4]{L} \div \sqrt[3]{S}.$$

When L is the length of ship in feet, and S the speed in knots, being the highest at which the ship can be driven without an excessive expenditure of power. That is, with a length of ship L,

$$\text{The maximum economic speed } S = (0{\cdot}4 \sqrt[4]{L} \div F)^3.$$

In a general way the resistance will vary very closely with the square of the speed with ordinary ships, which are not driven at a speed higher than that given by this rule.

The least length of ship for a given speed and coefficient of fineness can also be ascertained, thus :—

$$\text{Minimum length } L = \left(\frac{\sqrt[3]{S} \times F}{0{\cdot}4}\right)^4.$$

Table V. gives the prism coefficient suitable to the length of a particular ship for a given speed ; that is to say, the actual displacement of the ship of a given length and for a certain speed, should not be greater than that given by multiplying the displacement of a prism of the same section as the midship section of the ship and the same length by the factor given.

For example,—A ship 300 feet long, for a speed of 15 knots, should have a displacement not greater than ·674 of the prism displacement—that is to say, she must have a coefficient of fineness of waterlines of 0·674.

To determine the power necessary to drive a ship at the required speed, the facts already stated must be ascertained—viz., general dimensions, displacement, area of midship section, and the wetted skin. The first investigation must be to discover the highest speed possible under these conditions, and that is not less than that required.

Having done this, a calculation should be made of the maximum total resistance R in pounds. The efficiency of the machinery and propeller must also be known, and if the general efficiency is E, and S the speed in feet per minute—

$$\text{Gross indicated horse-power} = \frac{R}{E} \times S \div 33{,}000.$$

The total resistance is made up of three components :—

(1) That due to the skin friction, called *frictional resistance.*

(2) That arising from the making of waves and eddies, called *residual resistance.*

(3) That due to the action of the propeller on the hull, called the *augmented resistance.*

In the case of a paddle steamer the velocity given to the water by the wheels is higher than that of the water flowing past it, and increases the skin friction both before the wheels when the water is flowing into, and said to be feeding the race, and abaft the wheels in the race. But with a screw steamer there is also the increased velocity caused by the feed, but a greater loss is due to the decrease in pressure at the stern, owing to the action of the screw pushing the water away. So great is this in bluff ships that the water flows into the space behind the stern on each side of the race, and so causes an eddy stream to follow the ship.

The following table has been calculated by Mr. Johns, of R.C.N. Constructors, as applicable to all modern ships with clean, fresh-painted bottoms, and may be used for estimating the net horse-power necessary for overcoming the skin resistance. The horse-power to overcome the residual resistance can be calculated by means of Taylor's formula (p. 21). The two results added together will give the total net horse-power called E.H.P. If the propulsive efficiency is 0·6, then—

$$\text{I.H.P.} = \text{E.H.P.} \div 0{\cdot}6.$$

TABLE III.—Coefficients for Computing Effective Horse-power required to overcome Skin Friction based on Mr. Froude's Constants, as given by Mr. A. W. Johns.

If S is the wetted surface in square feet, then
E.H.P. = f . S, where f has the values given below.

Speed in Knots.	Length of Ship in Feet.								
	100	150	200	250	300	350	400	450	500
25, .	·2516	·2477	·2458	·2444	·2434	·2415	·2415	·2407	·2399
24, .	·2242	·2207	·2190	·2178	·2169	·2160	·2152	·2145	·2138
23, .	·1988	·1957	·1942	·1931	·1923	·1916	·1908	·1902	·1895
22, .	·1753	·1726	·1713	·1703	·1696	·1690	·1683	·1677	·1672
21, .	·1537	·1514	·1502	·1494	·1487	·1481	·1476	·1471	·1466
20, .	·1340	·1319	·1308	·1301	·1296	·1291	·1286	·1281	·1277
19, .	·1159	·1141	·1132	·1126	·1121	·1171	·1112	·1108	·1105
18, .	·0995	·0979	·0972	.0966	·0962	·0958	·0955	·0951	·0948
17, .	·0846	·0833	·0827	·0822	·0819	·0815	·0812	·0810	·0807
16, .	·0713	·0702	·0697	·0693	·0690	·0687	·0685	·0682	·0680
15, .	·0594	·0585	·0580	·0577	·0575	·0573	·0570	·0568	·0567
14, .	·0489	·0481	·0478	·0475	·0473	·0471	·0469	·0468	·0466
13, .	·0397	·0390	·0387	·0385	·0384	·0382	·0381	·0379	·0378
12, .	·0315	·0312	·0309	·0308	·0307	·0305	·0304	·0303	·0302

In the above table skin friction is taken as varying as $V^{1\cdot825}$.

In Table IV. are the values of C in the old formula $\text{I.H.P.} = \dfrac{D^{\frac{2}{3}} \times S^3}{C}$ for ships whose length varies from 100 feet to 900 feet, and the proposed speed from 10 knots to 28 knots, and designed with a form suitable for the speed—that is, the fineness of the water-lines of the ship is such that the prismatic coefficient of displacement will be not greater than given by the formula $0{\cdot}4\ \sqrt[4]{L} \div \sqrt[3]{S}$.

If the ship is finer than determined by this criterion, the value of C may be increased somewhat; also in the case of a ship having engines of high efficiency, such as possessed by most turbines, and high-class reciprocators

TABLE IV.—VALUES OF COEFFICIENT C IN FORMULA $I.H.P = \frac{D^{\frac{2}{3}} \times S^3}{C}$ WHEN COEFFICIENT OF WATER LINES IS NOT GREATER THAN $0{\cdot}4 \frac{\sqrt{L}}{\sqrt[3]{S}}$.

Speed in Knots.	10	12	14	16	18	20	21	22	23	24	25	26	27	28
100 feet long, . .	219	206	196	188	181	..	..	..	..	..	..	..	..	..
150 ,, . .	244	229	217	209	201	193	190	188	185	182	180	..	..	..
200 ,, . .	261	245	233	224	215	208	204	201	198	195	193	191	188	186
250 ,, . .	277	260	247	236	228	219	216	213	210	207	205	202	199	197
300 ,, . .	289	272	258	247	238	230	226	223	220	216	214	211	208	206
350 ,, . .	301	282	269	257	247	240	235	232	228	225	222	219	216	213
400 ,, . .	311	292	277	266	256	248	243	240	236	232	230	227	224	221
450 ,, . .	321	301	286	274	263	255	250	247	243	240	237	234	231	228
500 ,, . .	330	308	293	281	271	262	257	253	250	246	243	240	237	234
550 ,, . .	..	316	301	288	277	268	263	258	256	252	249	245	242	239
600 ,, . .	..	323	307	294	282	274	268	264	261	257	254	250	246	245
650 ,, . .	..	330	314	301	289	279	274	271	267	263	260	256	253	249
700 ,, . .	..	..	320	306	294	284	280	276	272	268	265	261	258	254
750 ,, . .	..	..	325	311	299	289	284	280	276	272	269	265	261	258
800 ,, . .	..	..	330	317	304	294	289	285	280	277	273	270	266	263
850 ,, . .	..	..	..	321	308	299	293	289	285	281	278	274	270	267
900 ,, . .	..	..	..	326	312	303	298	294	289	285	282	278	274	271

TABLE V.—Prism Coefficients of Ships suitable for Speeds and Lengths.

Length in Feet.	Speed in Knots per Hour.																					
	10·0	10·5	11·0	11·5	12·0	12·5	13·0	14·0	15·0	16·0	17·0	18·0	19·0	20·0	21·0	22·0	23·0	24·0	25·0	26·0	27·0	28·0
100	·590	·578	·568	·560	·552	·545	·539	·525	·513	·501	·491	·483	..	..	..	..	..	..	..	..	..	..
125	·623	·611	·601	·592	·584	·576	·569	·554	·542	·531	·520	·509	·501	·492	·484	..	..	..	..	..	..	..
150	·651	·640	·629	·620	·611	·603	·595	·580	·567	·555	·544	·534	·525	·516	·507	·499	·492	·486	..	..	..	..
175	·677	·664	·654	·645	·636	·626	·618	·603	·589	·577	·566	·554	·545	·536	·528	·520	·512	·505	·500	·492	·485	..
200	·698	·687	·678	·667	·657	·648	·640	·624	·609	·596	·584	·574	·564	·553	·544	·535	·529	·522	·515	·508	·501	·495
225	·718	·706	·696	·687	·676	·666	·658	·643	·628	·615	·602	·590	·580	·571	·562	·553	·542	·534	·527	·520	·513	·506
250	·738	·725	·714	·702	·693	·685	·675	·659	·641	·630	·618	·607	·595	·585	·576	·568	·560	·552	·544	·537	·530	·523
275	·755	·742	·731	·720	·710	·700	·692	·674	·659	·646	·632	·620	·619	·599	·588	·579	·573	·563	·557	·550	·543	·536
300	·772	·760	·740	·737	·727	·716	·707	·690	·674	·660	·647	·634	·623	·613	·603	·594	·586	·578	·570	·563	·555	·548
325	·784	·773	·762	·752	·741	·731	·720	·703	·689	·672	·660	·648	·636	·626	·615	·606	·598	·590	·582	·574	·566	·559
350	·802	·790	·779	·769	·759	·748	·739	·720	·704	·684	·675	·663	·648	·640	·627	·618	·609	·601	·592	·584	·577	·570
375	·815	·803	·791	·780	·769	·759	·748	·730	·713	·697	·684	·671	·659	·650	·636	·627	·619	·611	·603	·595	·587	·580
400	·829	·816	·804	·792	·781	·771	·760	·743	·725	·709	·696	·681	·669	·658	·647	·638	·629	·621	·613	·604	·596	·590
450	·853	·840	·828	·816	·804	·793	·783	·764	·746	·731	·716	·702	·690	·678	·667	·656	·648	·640	·631	·622	·614	·601
500	·878	·863	·850	·837	·826	·815	·804	·784	·767	·750	·735	·721	·708	·697	·687	·677	·666	·657	·648	·640	·631	·620
550	·899	·885	·870	·858	·846	·834	·823	·805	·785	·768	·754	·739	·726	·713	·702	·693	·682	·672	·663	·654	·645	·637
600	·918	·904	·891	·877	·864	·853	·841	·821	·801	·785	·770	·756	·742	·729	·717	·707	·697	·687	·678	·670	·660	·650
650	·937	·923	·909	·896	·882	·871	·860	·838	·820	·801	·786	·771	·758	·743	·732	·720	·711	·701	·692	·682	·673	·664
700	·952	·940	·926	·910	·898	·886	·871	·855	·832	·816	·800	·785	·770	·759	·745	·734	·724	·714	·704	·695	·685	·677
750	..	·955	·942	·925	·912	·901	·890	·870	·847	·830	·814	·800	·783	·772	·758	·743	·736	·726	·716	·707	·697	·688
800	..	..	·959	·941	·929	·917	·902	·883	·861	·844	·828	·812	·800	·785	·771	·760	·750	·739	·729	·719	·709	·700
850	..	..	..	·956	·943	·931	·915	·900	·874	·857	·840	·824	·810	·797	·782	·771	·761	·750	·740	·729	·720	·710
900	..	..	..	..	·957	·945	·930	·910	·887	·870	·853	·837	·821	·809	·796	·783	·772	·761	·750	·740	·731	·721
950	..	..	..	..	..	·957	·940	·921	·900	·881	·864	·848	·832	·819	·805	·793	·782	·771	·760	·750	·740	·730
1,000	..	..	..	..	..	..	·954	·934	·912	·900	·876	·860	·843	·831	·816	·805	·793	·782	·770	·760	·750	·740

having all the pumps disconnected from the main engines, and with efficient propellers C, may be somewhat higher. In the latter case the increase will be generally about 5 to $7\frac{1}{2}$ per cent. If, on the other hand, the efficiency of the machinery is from any cause low, as it used to be with the horizontal engines, and even with some of the vertical ones with air, circulating and feed pumps driven by the main engine; and if the efficiency of the screw is for some reason low, then the values given in the table are rather too high.

Further, it may not be assumed that such coefficients will be developed from the trials of such ships designed for and fitted with engines to drive them at high speeds when running at low; for example, it is a common experience to find the highest value at a speed 10 to 20 per cent. below the highest trial speed, and a decrease in value with the decrease of speed below; this is due to the fall in efficiency of the engines and propeller, both being too large for the power developed; as a matter of fact, the value of C is a measure of the general efficiency of a ship.

However, the calculations of resistance and net horse-power is scarcely the province of the engineer, and even the naval architect has found that experiments with models of ships in a tank are the most reliable way of ascertaining resistance and power. The Admiralty have employed a tank and trained staff of experimenters for more than 30 years, first under the guidance of the late Dr. W. Froude, since under that of his gifted son, Dr. R. E. Froude. Until recently tanks were a luxury enjoyed only by a few large wealthy shipbuilding companies; thanks, however, to the munificence of the eminent engineer, Sir A. F. Yarrow, there is now, at Bushey, a tank equipped with the very best apparatus open to all who desire to have experiments made with the models of proposed ships.

Tank Experiments with models of ships are very interesting, and of great importance to the builders of vessels out of the common order of things as to form and speed, as it is only by such means that their exact resistance under varying conditions can be computed with such accuracy and reliance as to permit of an exact provision being made of the power for the propulsion of the ships such models represent. In this way the designers of cruisers, scouts, destroyers, etc., whose form is uncommon and speed high, can determine the horse-power necessary for them, as also that for the very high-speed express steamers now required for service on channels and oceans.

Dr. William Froude's experiments with H.M.S. "Greyhound" and her model led to his establishing the laws which govern the true relation between ships generally and their models, as also those between one ship and another ship whose forms are similar but their dimensions different. Dr. R. E. Froude has for many years followed on with the work begun by his father, and from time to time has published his investigations and their results by reading papers at the Annual Meetings of the Institution of Naval Architects, in whose transactions they may be found recorded, and read with advantage. The fuller consideration of the subject, however, is one outside the scope of this work, except to say that Dr. R. E. Froude in this country, and Mr. Taylor in America, have developed methods whereby screw propellers may be examined and tested by their models, and their efficiency measured, not only *per se*, but when working at the stern of a model of the ship for which the screw itself is intended. By such experiments the effect on the ship of the screw working astern of it is also ascertained—that is, the *augmented resistance* due to the screw.

The tank is a canal wide enough, generally about 20 feet, and deep enough for such models as are used to pass through it without abnormal resistance; the model itself is made of paraffin wax to a suitable scale, and towed by mechanical means at a speed given by the following formula, when L and S are the length and speed of the proposed ship, and l and s that of the model:—

$$\text{Then speed of model } s = \sqrt{\frac{l}{L}} \times S.$$

If the model is made to the scale of a quarter of an inch to the foot, it will be one forty-eighth of the length of the ship, and consequently the speed practically is one-seventh that of the ship itself. The towing apparatus is on a travelling platform athwart the tank, which is caused to move at the speed required by electrical driving gear. The tension on the tow rope is carefully gauged and registered automatically, and gives the resistance of the model when free from the screw. The screw is then fixed to an apparatus on the same platform in rear of the model, and submerged so as to come into the exact position relative to the model that the real screw would be to the real ship. It is caused to revolve at the rate of revolution due to the speed of the ship and designed slip, but without propelling or even touching the model; its thrust is carefully measured, and the *torque* or power necessary to turn it also noted. The tension on the tow rope under these new conditions is also recorded, and compared with the preliminary records. The model is also tried in the same way at lower speeds, so that it has progressive trials similar to those the ship will or may have. By comparing the *thrust* with the *torque* at each speed, the screw's efficiency is ascertained, and can be plotted on a curve; by doing the same by the *tension* of the tow line and *torque*, the general efficiency of the ship can be compared in the same way, and by referring to the tests without the screw the augment of resistance due to the screw can be determined. In the case of the tension and thrust the speed in feet per second or minute is used as a multiple, while in that of the torque $2\pi \times$ revolutions in the same time is the multiplier to give the power usefully employed and that developed. The thrust multiplied by the speed is the measure of the screw as a "pusher"; the tow rope tension multiplied by the same speed is the useful work done, and, therefore, the measure of the screw as a propeller. Let T_r be the tow rope tension, T the screw thrust, and t the torque, R the revolutions, and s the speed in feet for time unit. Then

$$\text{Efficiency of screw} = \frac{T \times s}{6{\cdot}28\ R \times t}.$$

$$\text{Efficiency of propulsion} = \frac{T_r \times s}{T \times s} \text{ or } \frac{T_r}{T}.$$

For engineers' use there are several rules, which may be employed with advantage, and which will give the indicated horse-power under normal conditions with a fair amount of accuracy.

(1) **Professor Rankine's Rule** may be mentioned, although its employment is restricted; it is as follows:—

Rule I.—Given the intended speed of a ship in knots; to find the least length of the *after-body* necessary, in order that the resistance may not increase faster than the square of the speed: take *three-eighths* of the square of the speed in knots for the length in feet. To fulfil the same condition, the *fore-body* should not be shorter than the length of the after-body given by the preceding rule, and may with advantage be one and a half times as long.

Rule II.—To find the greatest speed in knots suited to a given length of after-body in feet, take the square root of two and two-third times that length.

Rule III.—When the speed does not exceed the limit given by Rule II., to find the probable resistance in lbs.: measure the *mean immersed girth* of the ship on her body plan; multiply it by her length on the water-line; then multiply by 1 + 4 (mean square of sines of angles of obliquity of stream lines). The product is called the *augmented surface.* Then multiply the augmented surface in square feet by the square of the speed in knots, and by a constant coefficient; the product will be the probable resistance in lbs.

Coefficient for clean painted iron vessels, . .	0·01
,, ,, coppered vessels, . .	0·009 to 0·008
,, moderately rough iron vessels, .	0·011 and upwards.

Rule III*a*.—For an approximate value of the resistance in well-designed steamers, with clean painted bottoms, multiply the square of the speed in knots by the square of the cube root of the displacement in tons. For different types of steamers the resistance ranges from 0·8 to 1·5 of that given by the preceding calculation.

Rule IV.—To estimate the *net* or *effective horse-power* expended in propelling the vessel, multiply the resistance by the speed in knots, and divide by 326.

Rule IV*a*.—To estimate the *gross* or *indicated horse-power* required, divide the same product by 326, and by the combined *efficiency* of engine and propeller In ordinary cases that efficiency is from 0·6 to 0·625 (Rankine, *Rules and Tables*). Marine engines to-day have a combined efficiency of 0·65 to 0·70, and some even higher.

Although the method here proposed has been found to give much more accurate and reliable results than those obtained by the older plans, it is open in practice to two very strong objections. First, it is necessary to have an accurate plan of the ship from which to measure the dimensions required; and second, it is difficult in actual practice to measure accurately the angles of obliquity of stream lines, and the calculation requires more time than can generally be devoted to the purpose. Often the horse-power requisite to drive a ship at a certain speed must be calculated at the time the lines are being got out, and it would be too late to wait for a plan of the ship before getting some idea of the power. Again, the size and fineness of a ship cannot be finally decided upon until the weight of machinery is roughly known; and as this will depend on the power, it is necessary to approximate to it on very rough and ready information. for which rough and ready rules are more suitable than the more refined ones. Hence, the rules based on immersed midship section and displacement could be conveniently used to obtain that approximation, and the power calculated accurately from the augmented surface afterwards.

Dr. Kirk's Analysis.—A method of analysing the forms of ships, and calculating the Indicated Horse-Power, was devised by the late Dr. A. C. Kirk, of Glasgow, and met with much favour on all sides. It is often used by shipbuilders on the Clyde and elsewhere for comparing the results obtained from steamers with those obtained from others, and likewise to judge of the form and dimensions of a proposed steamer for a certain speed and power.

The general idea proposed by him is to reduce all ships to so definite and simple a form that they may be easily compared; and the magnitude of certain features of this form shall determine the suitability of the ship for speed, etc. As rectangles and triangles are the simplest forms of figure, and more easily compared than surfaces enclosed by curves, so the form chosen by him is bounded by triangles and rectangles.

The form consists of a middle-body, which is a rectangular parallelepiped, and the fore-body and after-body prisms having isosceles triangle for bases; in other words, it is a vessel having a rectangular midship section, parallel middle body, and wedge-shaped ends, as shown in fig. 10.

This he called a *block model*, and is such that its length is equal to that of

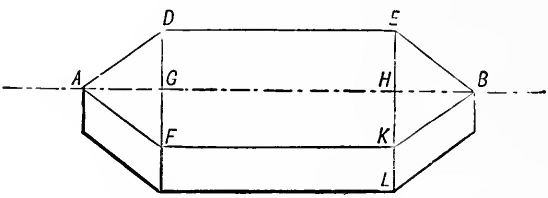


Fig. 10.—Kirk's Analysis.

the ship, the depth is equal to the mean draught of water, the capacity equal to the displacement, and its area of section equal to the area of immersed midship section of the ship. The dimensions of the block model may be obtained by the following methods:—

Since A G is supposed equal to H B, and D F equals E K, the triangle A D F equals the triangle E B K, and they together will equal the rectangle whose base is D F and height A G. Therefore, the area A D E B K F equals E K $\times$ A H. The volume of the figure is this area multiplied by the height K L. Then the volume of the block is equal to K L $\times$ E K $\times$ A H. But K L $\times$ E K is equal to the area of mid section, which is by supposition equal to the area of immersed midship section of the ship, and the volume of the block is equal to the volume displaced by the ship. Hence,

$$\text{Displacement} \times 35 = \text{immersed midship section} \times \text{A H};$$

Or,

$$\text{A H} = \text{displacement} \times 35 \div \text{immersed midship section.}$$

Now

$$\text{H B} = \text{A B} - \text{A H, and A B} = \text{the length of the ship.}$$

Therefore, the length of fore-body of block model is equal to the length of the ship, less the value of A H as found above.

Again, the area of section K L × E K is equal to the area of immersed midship section, and K L is equal to the mean draught of water. Therefore,

E K = immersed midship section ÷ mean draught of water.

Dr. Kirk also found that the wetted surface of this block model is very nearly equal to that of the ship; and as its area is easily calculated from the model, it is a very convenient and simple way of obtaining the wetted skin. In actual practice, the skin of the model is from 2 to 5 per cent. in excess of that of the wetted skin of the ship; for all purposes of comparison and general calculation, it is sufficient to take the surface of the model.

The area of bottom of this model = E K × A H.

The area of sides = 2 × F K × K L = 2 (A B − 2 H B) × K L = 2 (Length of ship — 2 length of fore-body) × mean draught of water.

The area of sides of ends $= 4 \times KB \times KL = 4\sqrt{HB^2 + HK^2} \times KL = 4\sqrt{\text{Length fore-body}^2 + \text{half breadth of model}^2} \times$ mean draught of water.

The angle of entrance is E B L; E B H is half that angle; and the tangent E B H = E H ÷ H B.

Or, tangent of half the angle of entrance = half the breadth of model ÷ length of fore-body.

From this, by means of a table of natural tangents, the angle of entrance may be obtained.

The block model for ocean-going merchant steamers, whose speed is from 15 knots upwards, has an angle of entrance from 24 to 15 degrees, and a length of fore-body from 0·3 to 0·36 of the length.

For that of ocean-going steamers, whose speed is from 12 to 15 knots, the angle of entrance is from 30 to 24 degrees, and fore-body from 0·26 to 0·3.

Rule.—For angle of entrance of "block model"—

$$\text{Angle in degrees} = 70\frac{\sqrt[4]{L}}{S}.$$

L is the length of ship in feet, S is the speed in knots.

Dr. Kirk measured the length from the fore-side of stem to the aft-side of *body-post* on the water-line. This is an unnecessary refinement when screw steamers alone are being compared, as then the length may be taken as that "between perpendiculars." However, when small or moderate size screw steamers are being compared with paddle-wheel steamers, it may be necessary to measure in this way.

(2) **The old Admiralty rules are as follows*** :—

(*a*) Indicated horse-power $= D^{\frac{2}{3}} \times S^3 \div C$.

(*b*) ,, ,, = area immersed with section $\times S^2 \div K$.

* If D_1 be the displacement in pounds, S_1 the speed in feet per minute, R the resistance in foot-pounds per minute, A the constant, then

$$R = D_1^{\frac{2}{3}} \times S_1^2 \times A.$$

Multiply both sides of this equation by S_1, then

$$R \times S_1 = D_1^{\frac{2}{3}} \times S_1^3 \times A.$$

Now $R \times S_1$ is the work done in overcoming the resistance R, through a distance S_1, and is, therefore, the power required to propel D_1 at a speed S_1, and if B is the efficiency

D is the displacement in tons, S the speed in knots, C and K are coefficients determined from previous practice. Their value varies with the size of the ship and the speed—that is, C and K will be less for a long ship than a short ship, the speed being the same—and if the length is the same, the value will be greater at the slower speed than at the higher.

The above two rules were, for many years, the only ones used by shipbuilders in determining the necessary power for a given speed. Their partial accuracy depended on the fact that the wetted skin varies very nearly with the displacement in ships of somewhat similar form,* and that the proportions of steamships were such that the wetted skin varied nearly with the area of immersed section. Their usefulness depended on the information in the hands of the user, and on his discretion in choosing values for C and K. These rules are, in experienced hands, a good check on the newer methods, and can be used by themselves with fewer data than are required when rules based on wetted skin, etc., are employed. Actual values for C are given in Tables vii., viii., ix., etc., on pages 41 to 48, deduced from the performances of ships on trial trips made with every care; but in choosing values discretion must be exercised that the ship for which a calculation is to be made is somewhat similar in form, size, and speed, to the one whose constants are selected. The values on Table iv. are made to suit all conditions.

The value of C may be taken as approximately $= 140 \times \sqrt[4]{L} \div \sqrt[3]{S}\ v.$ Table iv. gives them calculated in this way.

(3) Horse-power by calculation from wetted skin.

This is a simple and efficacious method, and one giving very satisfactory results in practice. It is based on the assumption that so long as a ship is

* *Note.*—Let L be the length of edge of a cube just immersed, whose displacement is D and wetted surface W. Then

$$D = L^3 \text{ or } L = \sqrt[3]{D},$$

and

$$W = 5 \times L^2 = 5 \times (\sqrt[3]{D})^2.$$

That is, W varies as D.

---

of the machinery and propeller combined, so that B × I.H.P. is the effective horse-power employed in propelling, then

$$33{,}000\ (B \times \text{I.H.P.}) = D_1^{\frac{2}{3}} \times S_1^3 \times A$$

$$\text{I.H.P.} = (D_1^{\frac{2}{3}} \times S_1^3) \times \frac{A}{33{,}000\ B}.$$

Now, it is more convenient to express the displacement in tons and the speed in knots; so that if D and S be substituted for D_1 and S_1, D being equal to $D_1 \div 2240$, and $S = (S_1 \times 60) \div 6080 = S_1 \div 101{\cdot}33$, it involves the introduction of other constant quantities, which do not, therefore, alter the expression, so that the whole of these constants may be replaced by a single constant, C, which will express them. Therefore

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}.$$

D being the displacement in tons; S the speed in knots; and C the coefficient.

It was also supposed that the resistance would bear a direct relation to the area of section transverse to the direction of motion, as this would be the measure of the channel swept out by a ship; hence the following rule:—

$$\text{I.H.P.} = \frac{\text{area of immersed midship section} \times S^3}{K},$$

K being also a so-called constant, but really only a coefficient.

not over-driven—that is, the speed does not exceed that appropriate to her form—the power will vary as the cube of the speed with machinery whose efficiency is not less than 0·9, and propellers suitable to the conditions, both as to diameter and area of blade surface; an allowance of 5 I.H.P. for each 100 feet of wetted skin at 10 knots is a fair basis for calculation, and easily to be remembered. It is obtained by supposing that the resistance per square foot of clean painted bottom at that speed should not exceed 1 lb. Dr. Froude found, with the "Alert," having a coppered-bottom, it was about $1\frac{1}{4}$ lbs., but a modern ship with a smooth steel surface coated with varnish paints will not cause so much resistance as old copper sheathing on a wooden ship. The author has come to the conclusion, from a careful examination of the trial results of a large number of ships, that it is frequently less than 1 lb., and that 1 lb. is a fair all-round allowance. The resistance, then, is 100 lbs. for 100 square feet, at the velocity per minute of 101·3 feet.

$$\text{Net horse-power} = 100 \times 1013{\cdot}3 \div 33{,}000 = 3{\cdot}07.$$

Taking the net horse-power as 62 per cent. of the gross, the

$$\text{I.H.P.} = 5{\cdot}0 \text{ nearly.}$$

If the efficiency, as is the case nowadays with high-class reciprocating engines, is 70 per cent., then the

$$\text{I.H.P.} = 4{\cdot}386 \text{ per 100 square feet at 10 knots.}$$

The same remarks apply in this case as to the former, as to the variation in value assignable due to the influence of length on speed; hence a suitable value to each case may be calculated as follows:—

$$\text{Rate of I.H.P. per 100 feet wetted skin at 10 knots} = 8{\cdot}5 \sqrt[3]{S} \div \sqrt[4]{L}.$$

If, then, the allowance for 10-knot basis is Q, then

$$\text{Gross I.H.P.} = \left(\frac{S}{10}\right)^3 \times Q \text{ for a speed of S knots.}$$

For Example 1.—Find the I.H.P. required to drive a twin-screw steamer 500 feet long at a speed of 23 knots, whose wetted skin is 40,000 square feet.

Here $\quad 8{\cdot}5 \sqrt[3]{23} \div \sqrt[4]{500} = 5{\cdot}10.$

$$\text{Allowance for 23 knots} = \left(\frac{23}{10}\right)^3 \times 5{\cdot}1 = 62{\cdot}05.$$

$$\text{Total I.H.P.} = 62{\cdot}05 \times 400 = 24{,}820.$$

Example 2.—How much I.H.P. will be necessary to propel a steamer 300 feet long at a speed of 21 knots, the wetted skin being 13,500 square feet, and the displacement 1,950 tons.

In this case $\quad Q = 8{\cdot}5 \sqrt[3]{21} \div \sqrt[4]{300} = 5{\cdot}63.$

$$\text{Allowance for 21 knots} = \left(\frac{21}{10}\right)^3 \times 5{\cdot}63 = 52{\cdot}2.$$

(*a*) $\quad \text{Total I.H.P.} = 52{\cdot}2 \times 135 = 7{,}042.$

By Admiralty methods, $\quad C = 140 \times \sqrt[4]{300} \div \sqrt[3]{21} = 212.$

$$D^{\frac{2}{3}} = 156. \quad S^3 = 9{,}261.$$

(*b*) $\quad \text{Total I.H.P.} = \dfrac{156 \times 9{,}261}{212} = 6{,}814.$

Example 3.—A cargo steamer 400 feet long has a displacement of 7,500 tons, a wetted skin of 29,500 square feet, what power is required to propel her at 12 knots?

Here
$$Q = 8{\cdot}5\,\sqrt[3]{12} \div \sqrt[4]{400} = 4{\cdot}38.$$
$$C = 140 \times \sqrt[4]{400} \div \sqrt[3]{12} = 274.$$
$$D^{\frac{2}{3}} = 383. \quad S^3 = 1{,}728.$$
$$\text{Allowance for 12 knots} = \left(\frac{12}{10}\right)^3 \times 4{\cdot}38 = 7{\cdot}56.$$
(*a*)
$$\text{Total I.H.P.} = 295 \times 7{\cdot}56 = 2{,}235.$$
By Admiralty method—

(*b*)
$$\text{Total I.H.P.} = \frac{383 \times 1{,}728}{274} = 2{,}397.$$

Example 4.—A yacht 230 feet long is required to steam 15 knots, her displacement is 1,100 tons, and the wetted skin 8,800 square feet, what I.H.P. should she develop?

Here
$$Q = 8{\cdot}5\,\sqrt[3]{15} \div \sqrt[4]{230} = 5{\cdot}37.$$
$$C = 140 \times \sqrt[4]{230} \div \sqrt[3]{15} = 221.$$
$$D^{\frac{2}{3}} = 107. \quad S^3 = 3{,}375.$$
(*a*)
$$\text{Total I.H.P.} = 5{\cdot}37 \times \left(\frac{15}{10}\right)^3 \times 85 = 1{,}594.$$
(*b*) ,,
$$= \frac{107 \times 3{,}375}{221} = 1{,}631.$$

It must not be forgotten, however, that these rules for speed and horse-power apply only to ships working under trial trip conditions, and the results shown in the following schedules are from trials made under the same circumstances—viz., in practically smooth water, free from eddies and cross currents, and when the wind and weather are moderate; further, the ship herself is in the best trim and condition, her bottom clean and fresh-painted, and, finally, the water is deep enough to preclude the influence of the sea bed seriously affecting her progress. The latter condition is an important one, and has had only in late years the consideration it demands, although it was patent to every one having to do with steamships long ago that in shoal water there was a marked diminution in speed. Now, however, we know, from the careful observations of many nautical authorities, that the influence continues, though in a lesser degree, when the ship is in deeper water than was formerly thought necessary for successful trials.

These things being so, the following rule should be observed if it is intended that the ship shall be able to attain the legend speed under somewhat unfavourable conditions of wind, weather, and condition of wetted skin, there must be provided a margin of power beyond that sufficient under trial trip conditions. This margin should be one of *power*, and *not one of speed*, for by the latter method the conditions are by no means satisfactory. It was, and is still, a common practice for the owners of express steamers to specify a trial trip speed considerably in excess of the speed desired on service; the hull is, as a consequence, of much finer form, with the corresponding lack of capacity for carrying deadweight. Moreover, the power and consequent weight of machinery is far in excess of what is necessary to fulfil the real requirements of the service.

For example, suppose a cross-channel steamer is required to perform her service in fair average weather at 20 knots. The owners specify that the trial speed is to be 22 knots, their real intention being to have a 10 per cent. margin.

Now, as a concrete example, suppose she is 350 feet long × 40 feet beam × 12 feet draught water, having a displacement of 2,644 tons, the following table of comparison of her with what she might have been had a 20 per cent. margin of power been provided for contingencies. It will be seen that the latter ship has 100 tons greater displacement, and her machinery, with the 20 per cent. excess power, is 120 tons lighter, and the cost will be about £7,000 less.

	s.s. A, in accordance with Specified Requirements.	s.s. B, for same Service and Scheduled Speed.
Length, ... ft.,	350·0	350·0
Beam, ... ft.,	40·0	40·0
Mean draft water, ... ft.,	12·0	12·0
Prismatic coefficient, ...	0·601	0·623
Displacement, ... tons,	2,644	2,744
Area immersed mid section, . . sq. ft.,	440	440
Maximum I.H.P., ...	9,000	7,800
I.H.P. for 20 knots, ...	6,322	6,488
Maximum speed—Maximum I.H.P., . .	22·35	21·2
Weight of machinery, ... tons	900	780
Difference in cargo capacity, ...	..	220
Increased consumption of coal per 100 miles,	..	13 cwt.

It will be observed in examining the schedules that, whereas the merchant ships are generally somewhat over-driven on trial for their forms, the naval ships are often finer than demanded by the legend or even actual speed, consequently they will attain their legend full speed under somewhat unfavourable circumstances.

The high speed of torpedo boats and destroyers depends almost wholly on their lightness of both hull and machinery, which enables them to do with so small a displacement that they literally skim the water, and the resistance per square foot of wetted skin is consequently comparatively small. Unless small boats are made to float at a very light draught they cannot be driven at high speeds, and all experiments with fast river steamers on the Clyde and elsewhere have shown the decided advantage of light draught. The effect on such ships of the depth of water in which they move is also considerable, and very interesting data have been given already showing this.

Progressive Trials.—In modern ship-trials information is usually sought both as to the power required for the highest speeds and (what is equally important) for lower speeds, as such knowledge gives the means of gauging the efficiency of ship and engines, jointly and separately, and is useful in ship designing.

The system of examination is as follows:—Let P_1 P_2 P_3 be the power developed in obtaining the speeds S_1 S_2 S_3 in knots with R_1 R_2 R_3 revolutions per minute. Take a line A N as a base line (fig. 11); on it take points B, C, and D, so that A B, A C, A D are proportional to S_1 S_2 S_3; at the points B, C, D erect ordinates, B *b*, C *c*, D *d*, so that they are proportional to P_1 P_2 P_3. Through the points *b*, *c*, *d* draw a curve, which is called the *curve of power*, or *curve of I.H.P.*, and it is such that if an ordinate be drawn through any other point, X, on the line A N, the part X *x* intercepted will measure the power corresponding to the speed measured by A X. If the curve is accurately drawn, it will be found that it does not pass through the point A, but above A, at a distance A *a*; this would signify that when the engine was indicating the power measured by A *a* the ship would not move, and so

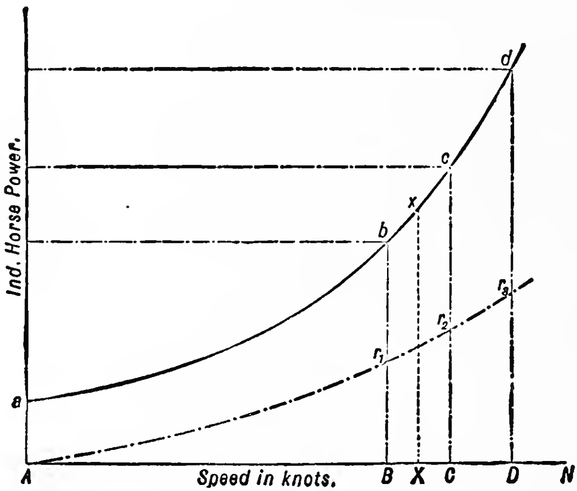


Fig. 11.

A *a* is the amount of power required to overcome the resistance of the machinery and propeller at starting, or rather, when not propelling the ship, and hence A *a* is said to represent the *initial friction* of the machinery.

Curve of Revolutions.—A curve of revolutions is constructed in a similar way, by taking points r_1 r_2 r_3 on the ordinates, so that Br_1, Cr_2, Dr_3 are proportional to R_1 R_2 R_3. When the slip is *constant* the curve of revolutions becomes a *straight* line.

Curve of Slip.—The slip may be shown by a curve whose ordinates are proportional to the percentage of slip at the speeds S_1 S_2 S_3 (*v.* fig. 12).

Examination of Curves will show—(1) what indicated horse-power revolutions and slip correspond to any speed intermediate to those observed; (2) the efficiency of the engine at its *lowest possible speed;* and from it an idea may be formed of its general efficiency, and a comparison made with other engines; (3) the efficiency of the ship, as tested by the rate of increase of power for speed, which is seen by the form of the curve towards the higher speeds—if it begins to mount upwards suddenly it is certain that the resistance has there begun to increase abnormally; (4) that, if the curve is one fairly following the law of resistance increasing as the square of the speed, an estimate may be made from it of the power requisite to drive a similar ship at speeds higher than the highest observed curves, or lower than the lowest; (5) any sudden rise in the slip alone indicates the propeller to be defective in either diameter or surface, or both.

Another method of expressing the results of progressive trials is by setting out A B, A C, A D proportional to S_1^3 S_2^3 S_3^3, and erecting ordinates, etc., as before. If the indicated horse-power throughout varies as the cube of the speed, the "power curve," or line drawn through the points *b*, *c*, *d*, will be a straight line; and if the power increases at a higher rate than the cube of the speed at any point, the line will again assume the curved form. The advantages of this plan over the one before described lie in the fact that a straight line is more easily drawn than any curve, that any deviation from a straight line is more easily detected than that of one curve from another, and that the production of a straight line is less liable to error than a curve, so that the interception of A*a* is less open to error than by the previous method. Of course, the curves of slip and revolutions cannot be examined so well by this latter method as by the former, and it is only the "power curve" that should be analysed in this way.

The values of the different "constants," rates, etc., for ships found from calculations made from the results of carefully conducted trial trips, are more reliable than those got by taking averages when employed in calculations for proposed ships. Tables vii. to xiii. give the values of constants, etc., as obtained from the performances of some well-known ships of various types and sizes.

Sea Performance of Steamers.—That the engines may work economically, both in consumption of coal and stores, as well as in wear and tear, it is advisable to run them at such a speed that they develop about 80 to 90 per cent. of the maximum power. Steamship owners do not always care to pay for 20 per cent. more power than is requisite to drive their ships at the speed intended, but there can be little doubt that it is true economy in the end to do so. For short voyages there is not the necessity for this reserve of power, and very fast steamers could not afford to carry the weight entailed by such an excess of power beyond the actual requirements; but ordinary sea-going steamers making long runs can, as a rule, easily do this without much sacrifice, and the wisdom of such a course would be shown by the saving in working expenses at the year's end.

Although, as a rule, trial trips are made honestly, and what the engine has done on the day of trial it can easily be made to do again, still, with the limited staff available for *continuous* service when the ship is at sea, there cannot be that attention devoted to the working parts which was bestowed by the staff of the manufacturer; and the application of water to the bearings and brasses to *prevent* heating, which is often deemed absolutely necessary by sea-going engineers when the engine is running at full speed, cannot but affect those parts prejudicially, and is a poor substitute for an attendant. Unless, then, the engine is run at a power somewhat below that of the trial trip, either a larger staff of engineers and attendants must be employed, or the wear and tear may be appreciable. It is true, on the other hand, that some engines will develop more power after a voyage or two than was obtained on their trial trip, due to the polishing of the rough surfaces of the guides and cylinder walls, and to the general "smoothing" of all the rubbing surfaces; but it is also true that even such engines should not be run for lengthened periods at their maximum power. The improvements in design and the employment of better materials for guides and bearings, however, admit of modern engines being worked at high speeds with less risk than was the case formerly. The better balancing of the engine with three cranks, and the almost perfect balancing of the four-crank engine, together with the extended use of superior white metal in guides and bearings, metallic packing in the principal stuffing boxes, have permitted of a high rate of revolution with less risk than obtained formerly with the slow-running mercantile engine; but, notwithstanding all this, a reserve of power for bad weather and emergencies is in every ship highly desirable.

Such remarks are, however, scarcely applicable to the turbine, inasmuch as it is cased in, and every provision made for high revolutions. Moreover, it is at high speeds that it is most economical, and in the mercantile marine is fitted to express steamers whose service demands full power. Moreover, it is claimed as one of the advantages of this form of motor that, in spite of high revolution, a smaller engine-room staff can be employed.

TABLE VI.—RELATION OF POWERS AND SPEEDS (SIR W. H. WHITE).

Description of Vessel.	Length in Feet.	Breadth in Feet.	Draught (Mean). Ft.	Draught (Mean). In.	Displacement in Tons.	I.H.P.'s and Admiralty Coefficients. 10 Knots. I.H.P.	10 Knots. Coeff.	14 Knots. I.H.P.	14 Knots. Coeff.	18 Knots. I.H.P.	18 Knots. Coeff.	20 Knots. I.H.P.	20 Knots. Coeff.
Torpedo Boat,	135	14	5	1	103	110	200	260	232	870	147	1,130	156
Torpedo Gunboat, "Sharpshooter" Class,	230	27	8	3	735	450	181	1100	203	2,500	190	3,500	186
3rd Class Cruiser, "Medusa,"	265	41	16	6	2,800	700	284	2100	259	6,400	181	10,000	159
2nd Class Cruiser, "Terpsichore,"	300	43	16	2	3,330	800	279	2400	255	6,000	217	9,000	198
1st Class Cruiser, "Edgar,"	360	60	23	9	7,390	1000	380	3000	347	7,500	295	11,000	276
1st Class Cruiser, "Blenheim,"	375	65	25	9	9,100	1500	290	4000	298	9,000	282	12,500	278
Atlantic Passenger Steamer,	525	63	21	3	11,550	2000	255	4600	304	10,000	297	14,500	281

TABLE VII.—RESULTS OF TRIALS OF STEAMSHIPS AT 9 TO 12 KNOTS.

DESIGNATION.	T.S.S. E—. E.	S.S.S. B—. A.	S.S.S. A—. O.	S.S.S. F—. A.	S.S.S. H—. S.	S.S.S. Cv—. D.	S.S.S. T—. E.	S.S.S. Bd—. A.	S.S.S. H—. A.	S.S.S. O—. O.	T.S.S. H—. R.	T.S.S. W—. D.	S.S.S. N—. E.
Length, perpendiculars, ft.,	120·0	200·0	330·0	250·0	255·0	310·0	319·0	320·0	392·0	420·0	460·0	469·0	426·0
Breadth, extreme, - ft.,	21·00	30·00	41·00	32·00	34·50	37·00	34·00	40·00	39·00	48·0	52·00	56·00	54·25
Draught of water, mean, ft.,	7·10	12·10	24 40	12·60	9·20	17·30	17·80	18·33	21·30	18·80	26·70	24·50	24·0
Displacement, - -tons,	282	1285	6880	1740	1420	3600	3445	4720	5767	8160	13,080	13,895	11,556
Midship section.immersed.area,	150	324	925	355	285	590	526	664	738	828	1,322	1,315	1,250
Kirk's Analysis. Wetted skin,- sq. ft.,	4508	8628	25,210	11,184	10,164	18,099	19 245	21,102	26,235	31,196	42,423	43,389	37,798
Kirk's Analysis. Length of fore-body, -	54·00	61·00	70·00	79·0	80·0	96·00	90·00	71·00	118·00	75·00	110·00	99·00	102·5
Kirk's Analysis. Angle of entrance, -	22° 0′	26° 30′	30° 20′	21° 36′	24° 0′	20° 6′	18° 38′	28° 38′	16° 30′	30° 0′	25° 21′	30° 20′	28° 27′
$0{\cdot}39\sqrt[4]{L} \div \sqrt[3]{S}$., - - -	·616	·664	·759	·678	·695	·721	·738	·734	·766	·780	·790	·792	·763
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, -	·550	·695	·788	·684	·700	·690	·712	·771	·698	·821	·753	·789	·760
Speed, - - - knots,	9·20	10·8	10·5	11·94	11·30	11·97	11·59	11·79	12·05	11·8	12·05	12·00	12·51
Ind. horse-power, - -	192	661	1372	962	816	1655	1194	1788	1758	2086	3382	3780	3305
I.H.P. per 100 ft. wetted skin, - - -	4·25	7·65	5·44	8·60	8·02	9·14	6·20	8·47	6·70	7·70	7·90	8·71	8·75
Do. reduced to 10 knots,	5·40	6·07	4·70	5·05	5·55	5·32	3·98	5·17	3·83	4·7	4·51	5·04	4·44
$\frac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, - - - - -	174	228	305	255	223	244	297	258	320	276	276	264	303
$\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, -	608	617	780	627	504	613	686	610	735	560	690	601	744
I.H.P. ÷ displacement, -	0·680	0·514	0·200	0·553	0·574	0·534	0·347	0·375	0·305	0·255	0·256	0· 272	0·286

TABLE VIII.—RESULTS OF TRIALS OF STEAMSHIPS AT 12 TO 15 KNOTS.

DESIGNATION.	S.S.S. A—. L.	S.S.S. R—. S.	S.S.S. Am—. H.	T.S.S. H—. D.	S.S.S. S—. N.	T.S.S. I—. Y.	S.S.S. L—. H.	T.S.S. A—. B.	S.S.S. C—. O.	S.S.S. B—. L.	S.S.S. B—. B.	T.S.S. D—. B.	S.S.S. A—. O.
Length, perpendiculars, ft.,	300·0	402·0	380·0	550·0	460·0	204·0	240·0	420·0	340·0	400·0	425·00	530·0	300·0
Breadth, extreme, ft.,	42·00	43·00	43·00	63·00	50·00	34·00	32·00	50·00	41·00	45·20	51·00	59·00	38·00
Draught of water, mean, ft.,	11·20	18·51	18·60	29·92	26·70	10·33	15·40	21·10	17·00	18·60	23·20	23·50	13·10
Displacement, tons,	2502	5670	5685	22,589	12,720	1080	2040	9070	4120	5842	9950	14,206	2506
Midship section, immersed, sq. ft.,	401	700	712	1,779	1,270	300	450	980	620	747	1126	1,225	453
Kirk's Analysis. Wetted skin, sq. ft.,	14,623	25,694	24,956	60,090	41,058	7929	12,022	33,065	20,128	26,063	36,448	46,391	14,594
Kirk's Analysis. Length, fore-body, ft.,	81·40	118·50	100·0	106·0	110·00	78·00	81·30	96·00	107·4	126·3	116·0	124·0	107·0
Kirk's Analysis. Angle of entrance,	24° 52′	18° 0′	21° 32′	31° 36′	24° 25′	20° 30′	20° 0′	27° 16′	19° 36′	18° 2′	23° 42′	22° 47′	18° 36′
$0{\cdot}39\sqrt[4]{L} \div \sqrt[3]{\text{Sp. ft.}}$,	·705	·757	·744	·808	·759	·623	·649	·734	·689	·711	·722	·764	·653
$\dfrac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$,	·729	·708	·737	·809	·761	·617	·661	·771	·684	·684	·727	·765	·643
Speed, knots,	12·23	12·28	12·4	12·75	12·95	13·20	13·23	13·90	14·36	14·79	14·8	14·7	15·33
Indicated horse-power,	1511	2101	2329	4790	3950	1018	1584	3908	2884	3742	4512	6118	2602
I.H.P. per 100 ft. wetted skin,	10·33	8·17	9·33	8·00	9·46	12·80	13·1	11·79	14·3	14·36	12·35	13·17	17·8
Do. reduced to 10 knots,	5·64	4·41	4·89	3·86	4·36	5·50	5·2	4·40	4·83	4·44	3·81	4·15	4·94
$\dfrac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$,	223	279	260	345	299	240	235	306	262	280	299	305	255
$\dfrac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$,	484	616	581	768	698	685	658	670	633	645	648	637	629
I.H.P. ÷ displacement,	0·604	0·371	0·410	0·213	0·31	0·942	0·776	0·430	0·700	0·645	0·452	0·430	1·04

TABLE IX.—RESULTS OF TRIALS OF STEAMSHIPS AT 15 TO 18 KNOTS.

DESIGNATION.	T.S.S. C—. R.	T.S.S. Pa—. R.	T.S.S. C—. D.	S.S.S. A—. N.	S.S.S. P—. A.	S.S.S. B—. C.	T.S.S. F—. B.	S.S.S. D—. N.	S.S.S. V—. A.	T.S.S. H—. D.	T.S.S. W—. F.	T.S.S. B—. T.
Length, perpendiculars, - - ft.,	280·0	290·0	300·0	313·5	420·0	450·0	220·0	388·0	466·0	600·0	680·0	570·0
Breadth, extreme, - - - ft.,	31·00	38·00	34·50	33·50	43·00	45·17	28·00	43·00	52·00	65·00	75·00	64·00
Draught of water, mean, - - ft.,	13·40	11·92	13·78	15·00	18·40	23·60	9·00	17·60	20·10	30·01	30·00	27·50
Displacement, - - - - tons,	1940	2100	2200	2480	5906	8500	736	4315	8124	23,630	30,020	19,150
Midship section, immersed area, -	350	416	425	422	698	926	204	615	980	1,810	2,094	1,629
Kirk's Analysis. Wetted skin, - - sq. ft.,	12,582	13,170	14,043	15,200	26,699	32,578	6841	22,330	28,511	63,977	76,440	56,195
Kirk's Analysis. Length of fore-body, - ft.,	86·00	113·3	129·0	107·5	124·0	129·0	94·00	142·0	176·0	143·0	178·0	168·0
Kirk's Analysis. Angle of entrance, - - -	17° 10′	17° 33′	15° 18′	14° 56′	17° 26′	17° 16′	14° 12′	14° 0′	15° 10′	24° 0′	22° 11′	20° 34′
$0·39 \sqrt{L} \div \sqrt[3]{Spd.}$, - - - -	·646	·653	·644	·660	·714	·728	·590	·676	·711	·766	·787	·737
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, - - -	·693	·609	·603	·658	·705	·714	·573	·632	·622	·762	·738	·723
Speed, - - - - - knots,	15·03	15·00	15·34	15·34	15·10	15·05	16·50	16·82	16·52	16·01	16·20	17·25
Indicated horse-power, - - -	2373	2157	2358	2243	4437	4900	1405	4660	6347	10,508	12,491	11,035
I.H.P. per 100 ft. wetted skin, -	18·6	16·38	16·78	14·75	16·60	15·04	20·40	20·87	22·26	16·41	16·35	19·57
Do. reduced to 10 knots, . -	5·47	4·85	4·66	4·09	4·83	4·42	4·55	4·40	4·94	4·00	3·85	3·81
$\frac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, - - - - - .	223	257	259	295	253	289	261	269	287	321	329	332
$\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, - - - -	501	650	650	678	543	642	654	626	696	707	712	760
I.H.P. ÷ displacement, - - -	1·22	1·03	1·07	0·90	0·75	0·58	1·90	1·08	0·78	0·44	0·42	0·57

TABLE X.—RESULTS OF TRIALS OF STEAMSHIPS AT 18 TO 21 KNOTS.

DESIGNATION.	T.S.S. Pe—. D.	T.S.S. Ch—. D.	T.S.S. Dr—. N.	T.S.S. Ly—. W.	T.S.S. Cl—. K.	T.S.S. Pg—. E.	T.S.S. H.M.S.P.	T.S.S. H.M.S.E.	T.S.S. C—. B.	T.S.S. B—. T.
Length, perpendiculars, . . ft.,	269·0	300·0	302·0	311·5	270·0	290·0	300·0	360·0	470·0	685·0
Breadth, extreme, . . . ft.,	33·00	34·50	38·00	36·00	34·00	38·00	36·50	60·00	58·00	68·00
Draught of water, mean, . . ft.,	8·75	13·78	13·50	11·50	10·50	11·92	13·90	20·60	20·50	29·91
Displacement, ... tons,	1,237	2,209	2,400	1,780	1,350	2,100	2,235	6,100	9,650	25,910
Midship section, immersed area, . .	266	425	460	392	311	416	445	1,059	1,100	1,922
Kirk's Analysis, Wetted skin, sq. ft.,	9,678	14,043	14,472	12,628	10,253	13,170	14,083	25,545	35,912	72,208
Kirk's Analysis, Length, fore-body, ft.,	106·3	129·00	119·4	152·5	118·0	113·3	124·0	158·0	163·0	213·0
Kirk's Analysis, Angle of entrance, .	16° 5′	15° 18′	16° 12′	12° 46′	14° 29′	17° 33′	14° 42′	15° 56′	18° 40′	17° 4′
$0·39 \sqrt[4]{L} \div \sqrt[3]{S}$., ...	·593	·617	·613	·617	·589	·590	·590	·616	·670	·725
$\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, . . .	·605	·603	·604	·514	·563	·609	·586	·561	·653	·690
Speed, ... knots,	18·9	18·20	18·57	18·70	19·3	20·34	20·8	20·96	20·20	20·8
Indicated horse-power, . . .	3,231	4,398	5,241	4,000	3,750	5,820	7,275	10,646	16,200	26,500
I.H.P. per 100 ft. wetted skin, . .	33·4	31·32	36·2	31·67	36·6	44·2	51·6	41·6	45·1	36·7
Do. reduced to 10 knots, . .	4·94	5·18	5·65	4·84	5·09	5·27	5·72	4·52	5·47	4·08
$\frac{\text{Disp.}^{\frac{2}{3}} \times S^3}{\text{I H.P.}}$, ...	246	232	219	240	207	236	212	288	230	297
$\frac{\text{Mid sec} \times S^3}{\text{I.H.P.}}$, ...	556	583	563	641	596	599	551	915	560	652
I.H.P. ÷ displacement, . . .	2·612	2·00	2·18	2·25	2·78	2·77	3·26	1·74	1·68	1·02

TABLE XI.—RESULTS OF TRIALS OF SCREW STEAMSHIPS AT 21 KNOTS AND UPWARDS.

DESIGNATION.	H.M.S. SN.	H.M.S. DV.	H.M.S. WF.	H.M.S. ATV.	H.M.S. PFR.	H.M.S. SKR.	H.M.S. CNL.	H.M.S. DRK.	T.S.S. CPA.	T.S.S. KWO.	T.S.S. DLD.	R.M.S. BTC.
Length, perpendiculars, . ft.,	200·0	210·0	210·0	374·0	370·0	360·0	440·0	500·0	600·0	678·0	663·0	852·5
Breadth, extreme, . . ft.,	19·5	20·5	21·7	38·25	38·75	40·0	66·0	71·0	65·2	72·0	67·0	94·0
Draught of water, mean, . ft.,	5·13	5·67	5·3	12·5	14·17	14·25	24·5	26·0	29·0	29·5	29·0	34·5
Displacement, . . . tons,	236	320	306	2,670	2,940	2,945	9,800	14,100	21,500	26,500	23,620	53.000
Midship section, immersed area, sq. ft.,	77·0	91·0	90·0	411·0	470	500	1,423	1,624	1,720	1,950	1,753	3,185
Wetted skin, . . sq ft.,	3,890	4,323	4,239	15,670	16,270	16,350	31,700	40,500	54,780	65,150	62,110	110,300
$0·4 \sqrt[4]{L} \div \sqrt[3]{S}$, ...	0·498	0·487	0·483	0·595	0·599	0·594	0·635	0·655	0·704	0·712	0·701	0·771
Prismatic coefficient, . . .	0·600	0·586	0·556	0·610	0·593	0·572	0·557	0·610	0·729	0·713	0·715	0·683
Speed, mean, . . . knots,	27·6	30·07	31·2	25·88	25·34	25·20	23·80	24·11	22·09	23·50	23·50	22·00
Indicated horse-power, . .	3,589	5,332	6,300	16,195	17,176	16,899	22,709	31,200	29,936	39,000	36,000	54,800
I.H.P. per square foot of wetted skin, ...	93·3	139·0	148·6	103·4	105·6	103·4	71·6	77·0	54·8	59·9	57·9	49·7
,, ,, reduced to 10 knots,	4·44	4·89	4·89	5·98	6·49	6·46	5·31	5·50	5·08	4·62	4·47	4·67
Displcmt.$^{\frac{2}{3}}$ $\times$ S^3 $\div$ I.H.P., .	237	239	219	206	195	202	272	262	278	207	297	288
I.H.P. $\div$ displcmt.$^{\frac{2}{3}}$, . . .	102	114	139	84·4	83·8	82·2	49·6	53·4	38·8	43·7	44·9	38·8

TABLE XII.—RESULTS OF TRIALS OF TURBINE SCREW STEAMSHIPS.

DESIGNATION.	TB.S.S. K E	U.S.A J.J.	TB S.S. QN.	TB S.S. B M.	H.M.S. AMST.	H.M.S. GRK.	H.M.S. SFT.	H.M.S. BLN.	H.M.S. GLS.	I.G.N. SDLZ.	H.M.S. INBL.	TB.S.S. LSTA.
Length, perpendiculars, . ft.,	250·0	310·0	310·0	375	360·0	255·0	345·0	385·0	430·0	656	530·0	760·0
Breadth, extreme, . . ft.,	30·0	29·75	40·0	46·0	40·0	25·7	34·2	41·5	47·0	93·5	78·5	87·50
Draught of water, mean, . ft.,	6·0	9·4	10·5	13·4	14·5	8·8	10·5	13·5	15·25	26·9	26·0	32·5
Displacement, . . . tons,	700	1,090	2,080	3,353	3,000	880	1,800	3,360	4,800	24,600	17,250	36,440
Mid section, immersed, . sq. ft.,	165	199	375	590	508	187	301	476	624	2,398	1,868	2,500
Wetted skin, . . sq. ft.,	6,400	8,710	12,200	18,830	16,066	7,580	12,000	18,100	22,250	67,000	47,340	82,350
$0\cdot4\ \sqrt[4]{L} \div \sqrt[3]{S}$, ...	0·588	0·543	0·602	0·609	0·604	0·513	0·524	0·585	0·613	0·665	0·637	0·714
Prismatic coefficient, . . .	0·573	0·616	0·625	0·530	0·574	0·632	0·609	0·640	0·625	0·545	0·660	0·671
Speed, mean, . . knots,	20·48	29·5	21·73	24·12	23·63	33·9	36·0	27·8	26·30	28·13	27·36	25·40
Shaft horse-power, . . .	3,500	18,625	8,000	14,700	14,200	14,250	30,000	18,000	25,417	89,738	47,300	64,600
S.H.P. per 100 square feet wetted skin, ...	54·7	213	65·6	78·0	88·4	186	250	99·4	114	134	100	78·4
" " reduced to 10 knots,	6·37	8·30	6·19	5·57	6·7	4·77	5·35	4·63	6·27	6·04	4·88	4·70
Displcmt.$^{\frac{2}{3}} \times S^3 \div$ S.H.P., . .	193	150	210	192	193·2	251	230	267	203	209	289	278
S.H.P. $\div$ displcmt.$^{\frac{2}{3}}$, . . .	44·4	175	49·1	65·6	68·3	155	203	80·4	89·2	36·6	70·8	58·7
Number of screws, . . .	3	2	3	3	3	3	4	4	4	4	4	4

TABLE XIII.—Results of Trials of Paddle-wheel Steamships.

Designation,	P.S. AMN.	P.S. JPN.	P.S. EGP.	P.S. JBN.	P.S. PCM.	P.S. LND.	P.S. VLT.	P.S. KNR.	P.S. LMG.	P.S. EMQN.	P.S. 1SCL.	P.S. LT2.
Engines,	Diag. Comp.	Vert. Oscil.	Vert. Oscil.	Diag. Comp.		Diag. Triple.	Vert. Triple.	Diag. Comp.	Diag. Comp	Diag. Comp.	Diag. Comp.	Diag. Comp.
Length, perpendiculars, ft.,	175	172·0	202·0	245·0	275·0	338·0	300·6	310·0	330·0	360·1	424·0	455·0
Beam, extreme, ft.,	26·0	18·8	21·0	29·0	34·8	34·75	33·0	32·0	40·0	42·3	52·6	55·3
Draught of water, mean, ft.,	3·00	6·80	6·25	6·10	11·00	9·00	10·67	5·75	8·75	13·0	12·50	13·13
Displacement, tons,	260	280	435	690	1,815	1,700	1,175	952	1,868	2,940	5,200	6,370
Mid section, immersed, sq. ft.,	74	99	122	160	360	292	287	175	322	466	600	690
Wetted skin, sq. ft.,	4,120	3,510	4,892	6,976	11,500	11,800	10,800	9,176	13,082	16,190	25,160	28,980
$0{\cdot}4\sqrt[4]{L} \div \sqrt[3]{S}$,	0·636	0·584	0·604	0·594	0·631	0·617	0·619	0·624	0·604	0·620	0·680	0·700
Prismatic coefficient,	0·700	0·576	0·614	0·616	0·642	0·607	0·600	0·620	0·615	0·623	0·710	0·709
Speed, mean, knots,	12·0	15·30	14·78	19·20	17·12	21·5	19·5	19·5	22·3	21·71	19·13	18·27
Indicated horse-power,	600	798	882	2,760	3,543	7,000	4,070	3,300	7,500	11,442	9,345	7,606
I.H.P. per 100 square feet wetted skin,	14·58	22·7	19·2	39·6	30·8	59·3	37·7	35·9	57·3	70·6	37·1	26·3
,, ,, reduced to 10 knots,	8·43	6·31	5·95	5·60	6·14	5·97	5·10	4·85	5·16	6·96	5·30	4·31
Displemt.$^{\frac{2}{3}} \times S^3 \div$ I.H.P.,	200	192	207	200	211	202	237	218	223	183	227	255
I.H.P. $\div$ Displemt.$^{\frac{2}{3}}$,	17·2	18·6	15·5	35·4	23·8	49·3	36·6	34·0	49·3	55·8	31·2	22·1

TABLE XIV.—RELATION OF POWERS AND DISPLACEMENTS.*

	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.
Length in feet,	280	300	360	435	500
Breadth in feet,	35	43	60	69	71
Mean draught in feet,	13	16½	23¾	24½	26¼
Displacement in tons,	1,800	3,400	7,400	11,000	14,200
I.H.P. for 20 knots,	6,000	9,000	11,000	14,000	15,500
I.H.P. per ton of displacement,	3·3	2·65	1·48	1·27	1·09

The following horse-powers were required to drive cruisers Nos. 4 and 5 in the above table at the speeds named:—

	No. 4.	No. 5.
10 knots,	1,500 I.H.P.	1,800 I.H.P.
12 ,,	2,500 ,,	3,100 ,,
14 ,,	4,000 ,,	5,000 ,,
16 ,,	6,000 ,,	7,500 ,,
18 ,,	9,000 ,,	11,000 ,,
20 ,,	14,000 ,,	15,500 ,,
22 ,,	23,000 ,,	23,000 ,,

The frictional resistance of clean painted surfaces varies about as the 1·83 power of the speed, but resistance due to wave making may vary very widely, since it is dependent on form. The total resistance of "Destroyers" has been found to vary as follows: *—

Up to 11 knots, - - -	nearly as speed^{2}
At 16 ,, - - -	,, speed^{3}
,, 18-20 ,, - - -	,, speed$^{3·3}$
,, 22 ,, - - -	,, speed$^{2·7}$
,, 25 ,, - - -	,, speed^{2}
,, 25-30 ,, - -	practically as speed$^{1·83}$

and the resistances other than frictional vary as follows:—

Up to 11 knots, - - -	as speed^{2}
At 12½ to 13 knots, - - -	,, speed^{3}
,, 14½ knots, - - -	,, speed^{4}
,, 18 ,, - - -	,, speed (more than 5th power)
,, 24 ,, - - -	,, speed^{2}

and at higher speeds as still lower powers of the speeds.

The relation of the frictional to the total resistance is *:—

	"Destroyer."	Cruiser.
At 12 knots,	80 per cent.	90 per cent.
,, 16 ,,	70 ,,	85 ,,
,, 20 ,,	nearly 50 ,,	nearly 80 ,,
,, 23 ,,	...	over 70 ,,
,, 30 ,,	45 ,,	...

If the coefficient of friction be doubled (as it might easily be with a foul bottom), the maximum speed of the "Destroyer" would fall fully 5 knots, and that of the cruiser would be reduced to 19 knots. See Tables vii. to xiv.

* Sir Wm. White, British Association Address, 1899.

Progressive Trials should be made with all ships when at the measured mile, and it should be remembered that for practical purposes it is more important to know the power, revolutions, slip of propeller, etc., at speeds

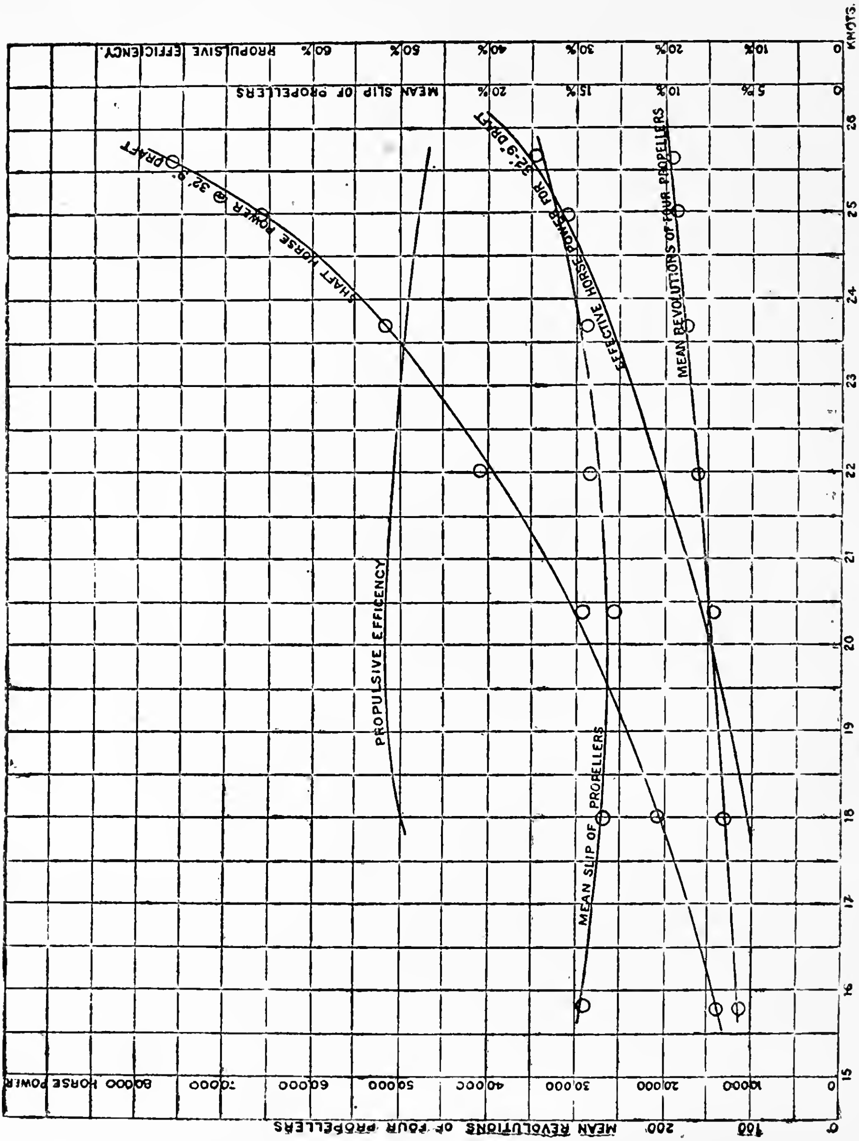


Fig. 12.—R.M.S. "Lusitania's" Progressive Trial off Skelmorlie.

less than the maximum than those at the utmost speed; it is especially important that the ship shall be tried at as low a speed as possible consistent

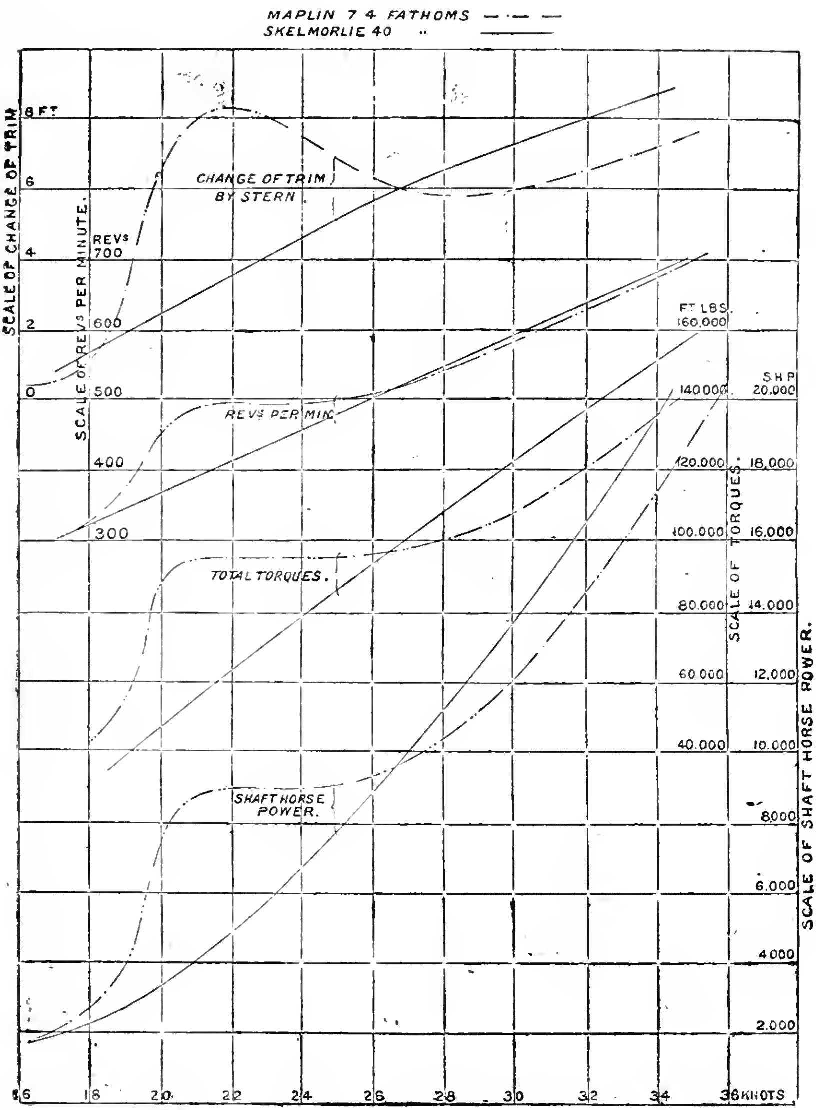


Fig. 13.—Effect of Depth of Water on Performance.

Speed Trials of H.M. Torpedo-boat Destroyer "Cossack" at Maplin and Skelmorlie. 270′× 26′× 9·3′ draught. Displacement, 836 tons. 14,000 S.H.P.

with obtaining accurate observations; and that between it and full speed there should be one or more trials at intermediate speeds. There should be three consecutive runs made on the mile for each rate of speed, with the steam, vacuum, and revolutions kept as steady as possible during the whole time occupied in doing them. The mean speed should be calculated in the usual way—that is, if the speeds observed are at the rate of x, y, and z knots per hour—

$$\text{True mean speed} = \frac{x + 2y + z}{4}.$$

The revolutions should be taken from a counter, as the number per mile,

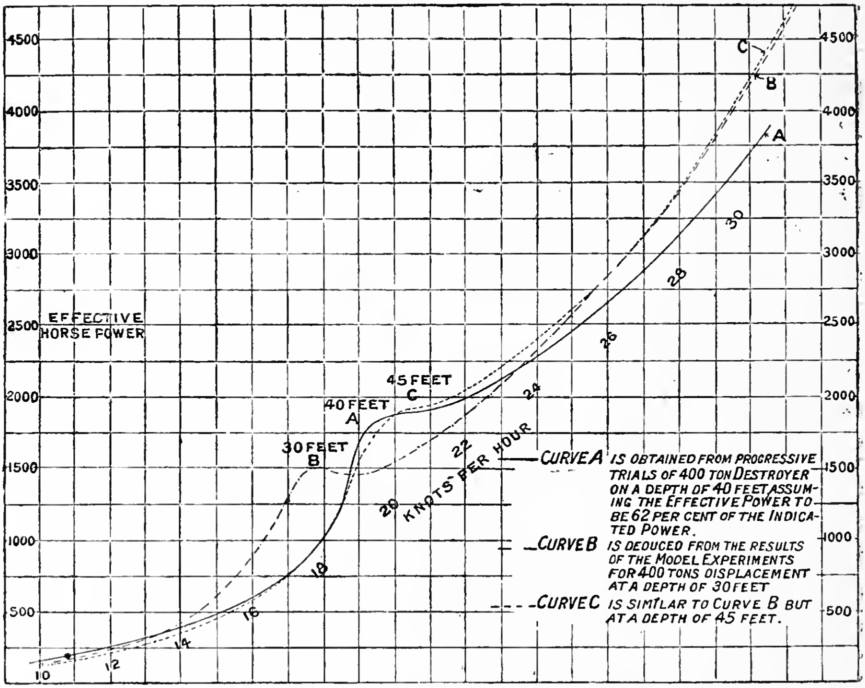


Fig. 14.—Harold Yarrow's Experiment.

Curves of Comparison between Model Experiments of 400 tons at depths of 30 and 45 feet, and Actual Trials with a Destroyer of 400 tons at a depth of water of 40 feet.

this, when divided by the time taken on it, will be the true rate of revolution per minute. The results of horse-power, revolutions, slip of screws, and value of $C = \frac{D^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$ as a measure of efficiency for each speed should be set up as ordinates, with the speed in knots per hour as *abscissæ:* a curve drawn through their upper ends of each set will indicate all that is desired to know of the ships' performance. If, however, the ship's resistance has been ascertained by model experiments or otherwise, a curve of E.H.P. can be inscribed, and the curve of propulsive efficiency will then be obtained by inscribing the values of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ as ordinates for it. Fig. 12 is such a diagram, as it is highly desirable to have for all important ships, showing, as it does, clearly the performance of the "Lusitania" on the measured mile. Being a turbine-driven ship, the power is practically Brake Horse-Power, being that obtained by observing the torque on the propeller shafting. It is usual to speak of this as Shaft Horse-Power.

The Effect of Depth of Water on the speed of steamships is somewhat erratic, as may be seen by carefully examining the curves of performance of several ships and models at trials made from time to time for this purpose.

Sir Philip Watts exhibited those of H.M.S. "Cossack," obtained as the

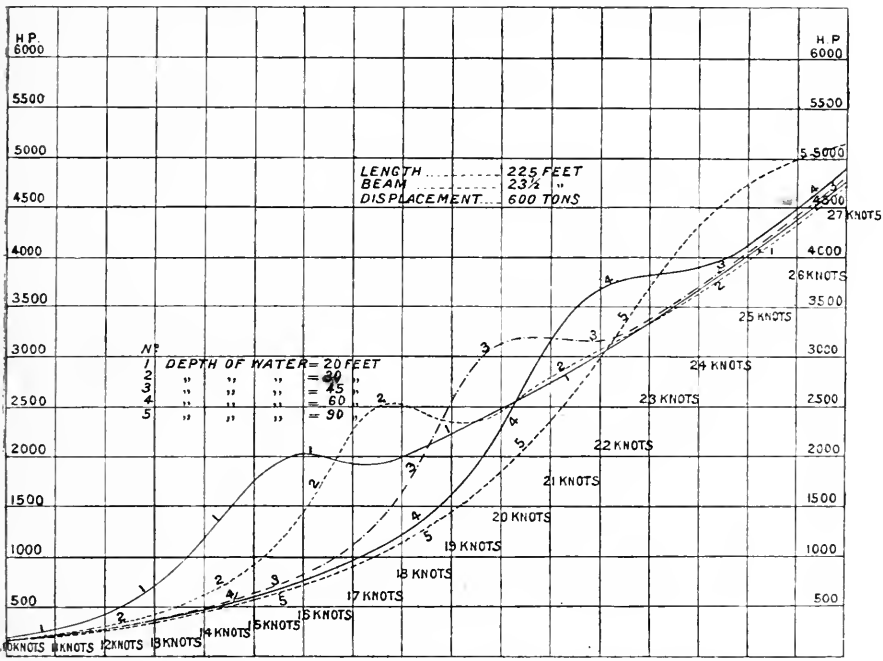


Fig 15.—Curve of Effective Horse-power and Speed with Various Depths of Water. Model Experiments by Harold Yarrow, I.N.A.

results of her trials at the Maplin Sands, where the water is comparatively shallow (7·4 fathoms), and those carried out at Skelmorlie, where the water is deep—viz., 40 fathoms—and the influence of the bottom only felt by the largest ships at high speeds. It will be seen that, at what may be called the critical speeds of this ship, there were changes of trim corresponding to the changes in revolution and torque at about 18 knots; these were of a violent nature at about 20 knots, while at 26·5 knots things became normal again; at speeds above this the power required was actually less in the shallow water than in the deep (*v.* fig. 13).

Figs. 14 and 15 are equally interesting, as being the results of special

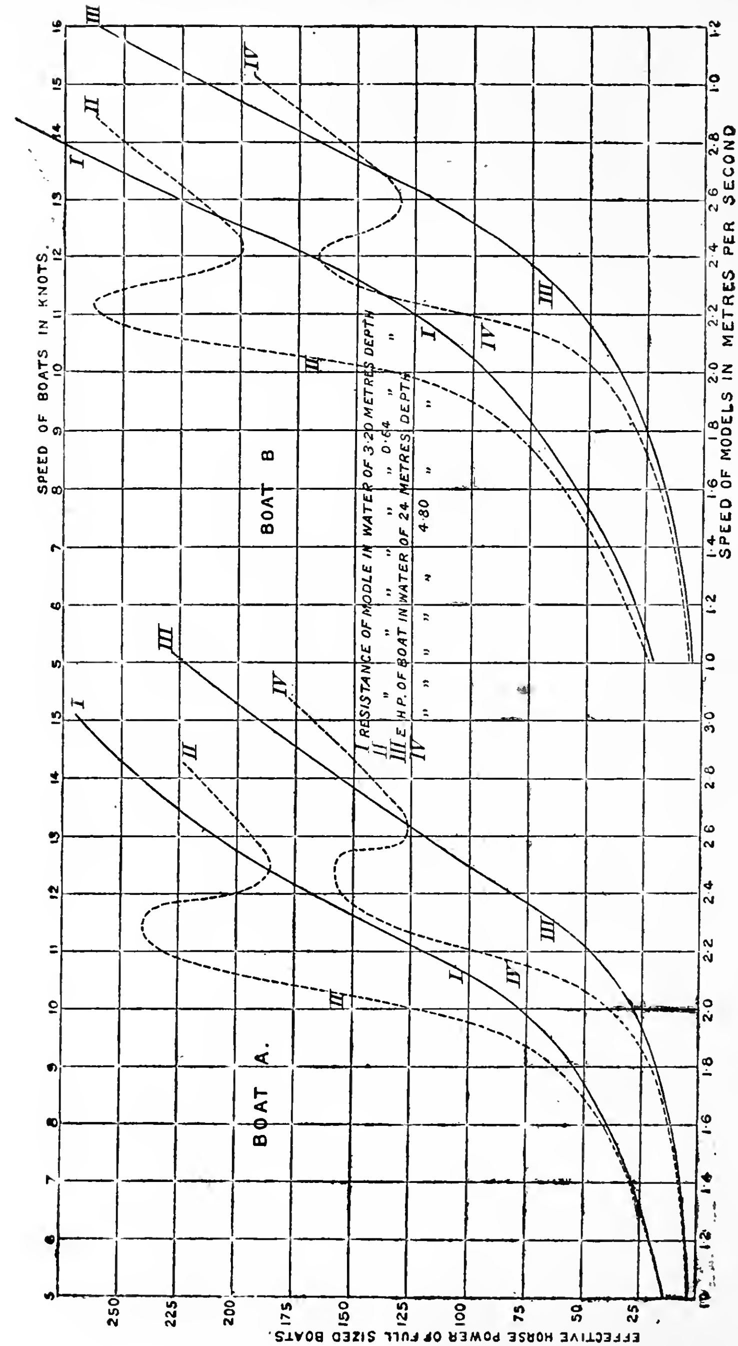


Fig. 16.—Model Experiments made by N. D. Lloyd, S.S. Co., Bremen (Herr Popper).

Boat A—73·5 feet × 11·5 feet × 2·56 feet draught, mean. Model, 9·8 feet long.

„ B—93·0 „ × 14·2 „ × 2·3 „ „ „ 12·4 „

trials made by Mr. Harold Yarrow for the same purpose of finding the effect of depth of water on fast ships. Fig. 14 shows the curves of comparison between the model experiments and those made with the actual ship. The ship herself acted in much the same way as did the "Cossack" in 7·4 fathoms; her critical point was 18 knots, but became normal at about 23 knots.

The North German Lloyd Company, of Bremen, had an interesting series of experiments made by Herr Popper, and fig. 16 shows the results of two sets of them, each being made with the boat and her model; the effects in both cases are even more striking than the former ones, inasmuch as the changes are more emphatic and pronounced.

The full accounts of all these trials are given in the *Transactions of the Institution of Naval Architects*, and may be studied there with advantage.

Dr. D. W. Taylor's formula for ascertaining the least depth of water in which a ship should undergo her speed trials for a satisfactory performance is as follows:—Minimum depth of water in fathoms $= \dfrac{10 \times d \times s}{l}$, where d is the draught of water of the ship, l its length between perpendiculars, and s the speed in knots.

Example.—The minimum depth of water for the trials of a cruiser whose draught is 26 feet, the length 500 feet, and the speed 25 knots.

$$\text{Depth} = \frac{10 \times 26 \times 25}{500} = 13 \text{ fathoms.}$$

CHAPTER III.

MARINE ENGINES: THEIR TYPES AND VARIATIONS OF DESIGN.

THE marine engineer, when dealing with design and construction, is faced with, and has to solve, problems much more complex and involved than those corresponding ones familiar to builders of land engines. Moreover, he is hampered by circumstances and limitations quite unknown to the latter.

The space occupied by, and the weight of, the machinery of the ship is limited at all times; in the case of the cargo ship each ton of weight means a ton less cargo on which freight is payable, and in that of the express steamer and warship where the power is larger in proportion to the size of the ship, both weight and space are of great consequence, and generally quite extremely limited. The design, therefore, must be such to allow of inclusion in the machinery space allotted, while leaving sufficient room to permit of accessibility to all parts as required for working and overhauling; the weight is as strictly limited to that share of the displacement provided by the naval architect in his design. In bygone years more than one good ship failed to comply with the conditions prescribed for her by her designers from a miscalculation of the weight of machinery, or the adding to it by the engine builders without regard to the consequences. To the fast paddle steamer it was fatal, for, in addition to the extra displacement and wetted skin, there was an increased immersion of float, which reduced the efficiency of the propeller immensely. The large engines found in central electric light and power stations on shore are secured on massive concrete foundations which never move, whereas those on board ship, while being strong, are not massive, and may and do move about in a way most trying to them and their bed-plates. Not only does the engine move forward with the ship in the line of the course she pursues, but it is liable to inertia stresses due to the acceleration and retardation of the velocity of the ship from various causes, and when pitching and rolling the angular motion is often considerable, so that the inertia stresses thus set up are even more serious. It has not often happened that a marine engine was torn away from its bed, but all of them are liable to such an accident if ample provision is not made for such a contingency.

In warships very special provision was formerly always made for the shocks that would be set up when ramming the enemy; this, however, is a nautical manœuvre no longer contemplated by naval commanders, as it was apt to be more fatal to the rammer than the rammed, so that for this and other reasons ramming is no longer spoken of. It remains, however, as a contingency common to all ships, for both naval and mercantile ships are liable to run on a rock or other massive obstruction, and even to ram another ship accidentally. It is not desirable that the displacement of the engines shall follow such a catastrophe.

The marine engineer has also to produce an engine that may be depended on to keep running without a stoppage for an indefinitely long time, and at a uniform speed, as it is highly desirable that there should be no slowing down of the engine except when so desired by those navigating the ship. Slowing down or stopping the engines at a critical moment might mean the loss of the ship; in the case of a naval ship, it might mean her capture, or even the loss of a battle, which would decide the fate of a kingdom. It is, therefore, of the very first and highest importance that the marine engine shall be free from every extraneous fitting which might cause temporary derangement, and itself should be so carefully designed, manufactured, fitted, and cared for as to preclude the possibility of compulsory stoppages.

Further, in both naval and mercantile ships, the whole of the machinery must be practically noiseless when in motion, and the engines free from vibration, which would spoil the gunnery of the one and the comfort of the passengers in the other; even the auxiliary machinery must comply with these conditions of absence of noise and vibration. Finally, since there is a limitation to the weight of fuel which can be carried, and it is desirable that it shall last over as long a voyage as possible, it is necessary on that ground, as well as for the sake of economy in cost, that the consumption of it be as low as possible consistent with a satisfactory compliance with the conditions already insisted on as essential.

The marine engineer enjoys one advantage over his brother on shore; he has an unlimited and cheap supply of cold water, whereby he may condense as much steam as he desires, and work with as high a vacuum as his apparatus can produce and maintain. And if it be an advantage, and no doubt it is in many ways, his engine varies in velocity as its load varies, instead of running at constant velocity whatever be the load.

Various Types and Designs of Engine, of which in the early days of steamships there was a very large variety employed, and although some of them were developments of those common and successful for land service, the greater number were evolved to satisfy the conditions imposed on the engineers of the day, many of them showing considerable originality in form as well as ability in design; some of them display features which indicate that their originators possessed a technical knowledge, for which they have not been always accorded credit. On the other hand, not a few had inherent defects, which, while not being apparent in the model state or in engines of small power, were very evident in the larger engines, and soon caused their early dismissal to the scrap heap; such defects were generally due to a want of technical knowledge, or to the misapplications of the empiric formulæ of the day, which were the rough and ready pilots of these earlier times.

The engineer of to-day is not required to waste time and energy in differentiating the claims of many types before coming to a conclusion as to what will suit the circumstances of his particular case, for time and experience have decided much of this for him, and so now for a paddle steamer it is almost the universal rule to choose the direct-acting compound engine, either horizontal or nearly so, and for the screw the inverted form of the same. He may, however, have to debate with himself or his advisors whether his is a case for turbines pure and simple or for reciprocators; he may also compromise the matter by having reciprocators at the boiler end of the installation, with low-pressure turbines at the condenser end, as is the case

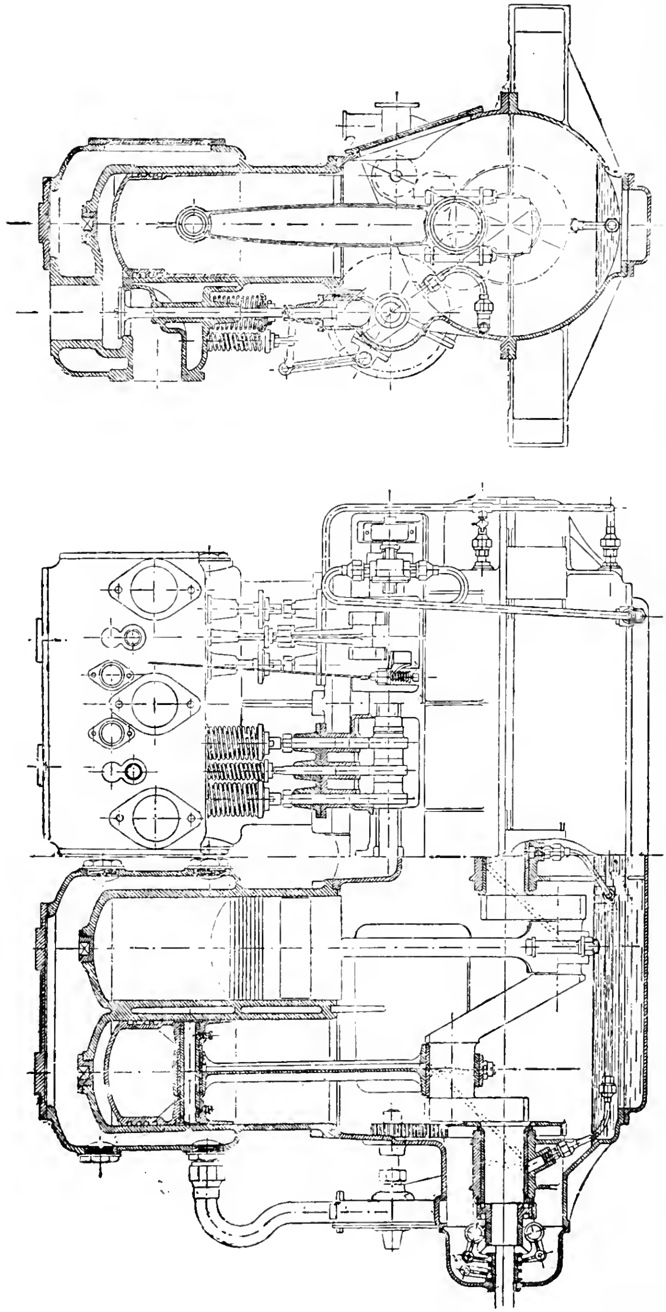


Fig. 17.—100 B.H.P. "Thornycroft" Four-cylinder, 8" × 8", Marine Oil Engine.

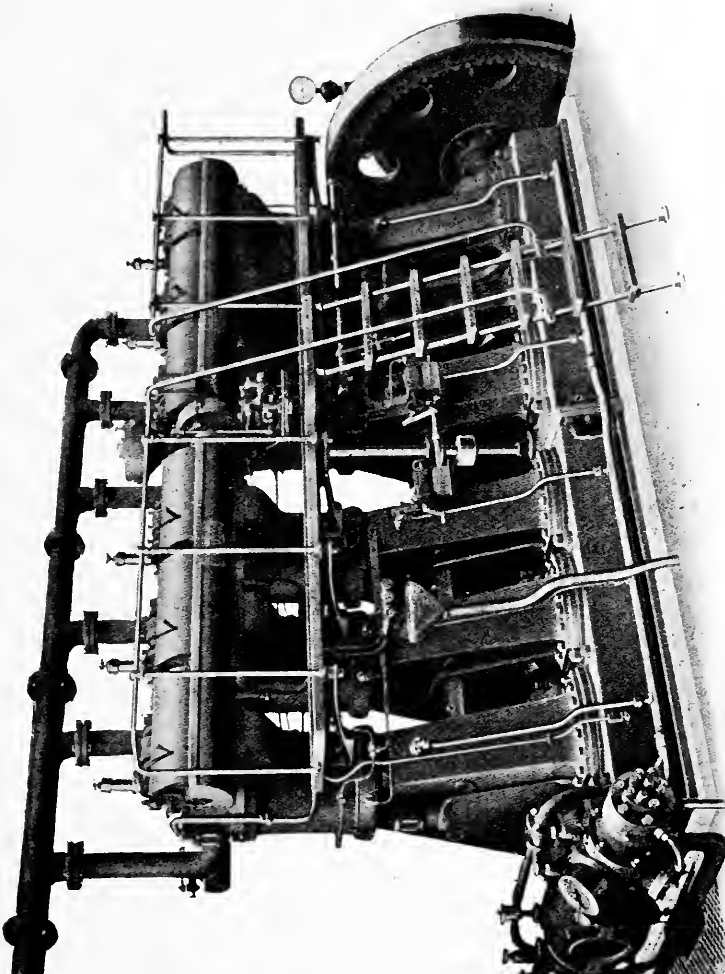


Fig. 18.—Six-cylinder Heavy Oil Engine. (Belliss and Morcom Four-cycle Single-acting—Diesel system.)

now in many large ocean-going ships of high speed (*v.* fig. 44). When the power is not very great a Diesel oil engine may be adopted, and with smaller power even a semi-Diesel type will prove attractive. For small craft of low or moderate speed the paraffin engine is popular and convenient. Petrol, however, should be avoided on shipboard except in very special cases.

Although there is very little variety in the type of engine now employed for paddle-wheel ships and screw ships, it is well to know why particular ones have survived, while others have died out, and in order to appreciate the selection, some knowledge of the virtues and vices of those others now discarded is desirable as well as instructive.

Paddle-wheels were the first practical form of propeller employed for steam navigation, and, as has already been stated, the earliest ship had a horizontal direct double-acting engine. To-day this form of engine survives as the fittest for the purpose of both side wheel and stern wheel paddle-ships of all sizes; in the case of the former it is, however, inclined to the horizontal position, and called the diagonal or inclined engine. In this form it was patented by Marc Isambard Brunel, the famous engineer, in 1822, who claimed as his invention two inclined cylinders, the cranks at right angles, the piston-rod ends fitted with roller guides, and the weight of the pistons relieved by spring supporters on the extremities of the beam head; the engines were to be governed by means of a stream of water pumped through an orifice. The condenser was to be formed of an assemblage of pipes, which collectively formed a spacious chamber; they were to be connected together with a set of smaller pipes, and the whole placed in an iron reservoir. This must be admitted to be a very ingenious and comprehensive invention, considering it included roller guides, balanced pistons, a cataract governor, and last, but not least, a surface condenser. The roller guide never took on for large engines, and a cataract governor seems almost to have met with the same fate, otherwise Brunel's invention may be said to have survived as the fittest for the purpose of marine propulsion with paddle-wheels, just as the vertical direct-acting inverted first adopted by the Thomsons, of Glasgow, has for screw propeller work.

The inclined direct-acting engine (figs. 19 to 22) is the engine of to-day, as being the one complying best with the conditions ruling on board such ships as are still propelled by side paddle-wheels for the following reasons :—

(1) The design is simple and without complications or make-shifts of any kind, hence less liable to derangement and breaks down.

(2) It is as light and cheap in construction as any other, and may be arranged so that its framing helps to stiffen a lightly built ship rather than to over-strain her.

(3) Great flexibility of design is possible; the engine may have as many cylinders as are desired within the limits of beam, and may be compound or simple without any difficulties or objections.

(4) The stroke of piston may be long; in fact, there is practically no reasonable limit to it.

(5) The steam pressure may be as high as that in screw ships without any disadvantages.

(6) The shaft may be at any desired height from the ship's floor, and so permit of a small diameter of wheel with a corresponding increase in number of revolutions if desired.

On the other hand, it has the defects of all horizontal engines, such as heavy pistons running on the cylinder liners or bodies either unsupported or inadequately so, whereby they are worn barrel shape in course of time; there is also the friction due to this which retards the movement. Further, the momentum of the pistons and rods in connection therewith causes a pulsation fore and aft-ways in the ship, which in light river craft is sometimes

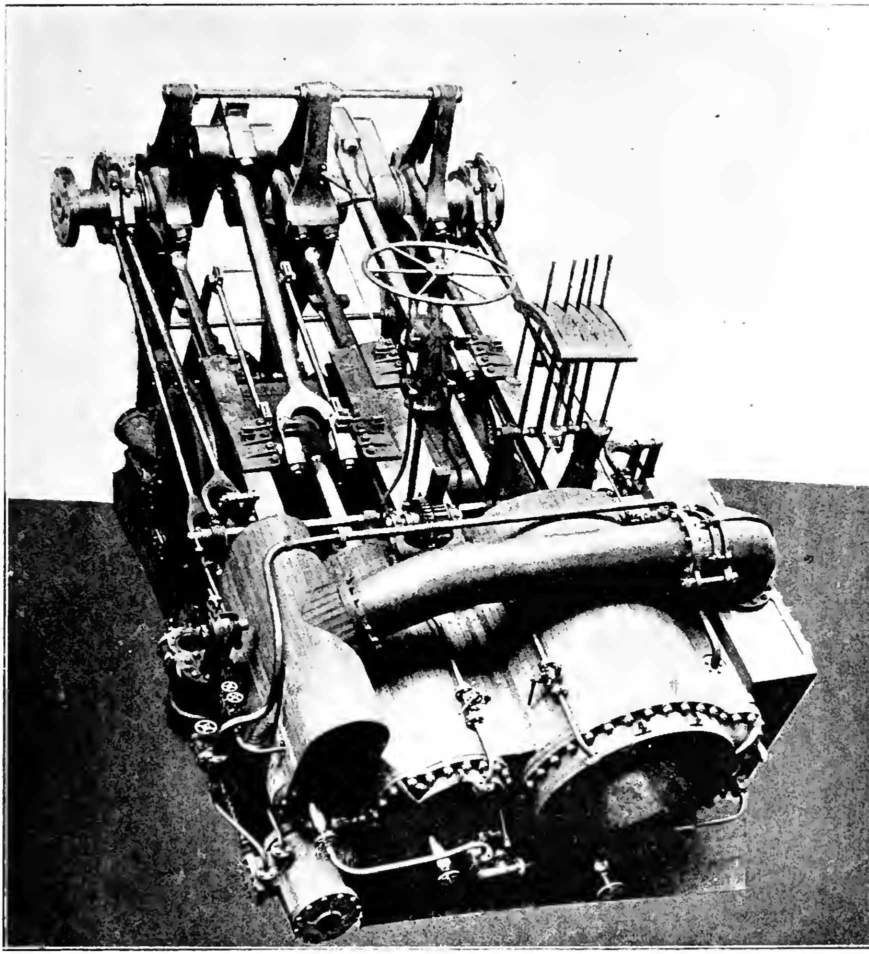


Fig. 19.—Diagonal Compound Paddle Engines (J. Brown & Co., Clydebank).

by no means agreeable to the passengers; this is especially the case in some of the older river steamers having only one cylinder, the more modern steamers having three cylinders (fig. 22) are practically free from this defect, and in those with two cranks at right angles it is not so serious, but it is yet observable, whereas with vertical cylinders it was absent, although they had a vibration of their own, and the steeple engines had the objectionable pulsation also. But, as a rule, the paddle steamer in old days was a more comfortable one in which to sleep than were the screws.

Of the inclined types, there may be one or more cylinders side by side, as shown in fig. 19; or, when the ship is narrow, and from this or other cause a limitation in athwartships space, the design of fig. 20 is a good one, and

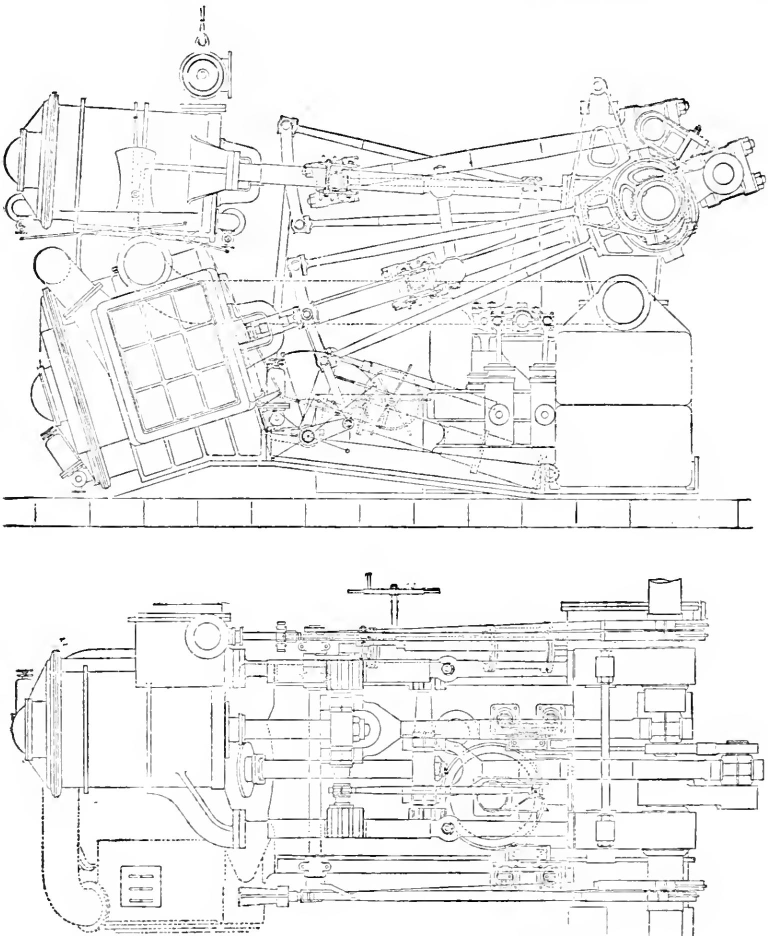


Fig. 20.—Diagonal Compound Paddle Engines.

works quite satisfactorily. This latter permits of more deck room amidships, and space available for passengers, etc., and generally gives a clear deck fore and aft in even small ships.

The fore and aft space taken up by the diagonal engine is of course very considerable, and formerly in some classes of ships would be of great disadvantage, but in modern paddle-steamers this is of no consequence, and is more than compensated for by the advantages in other ways, not the least of which is the ease with which a paddle-wheel of small diameter can be adopted, whereby high revolutions are rendered possible. This, together

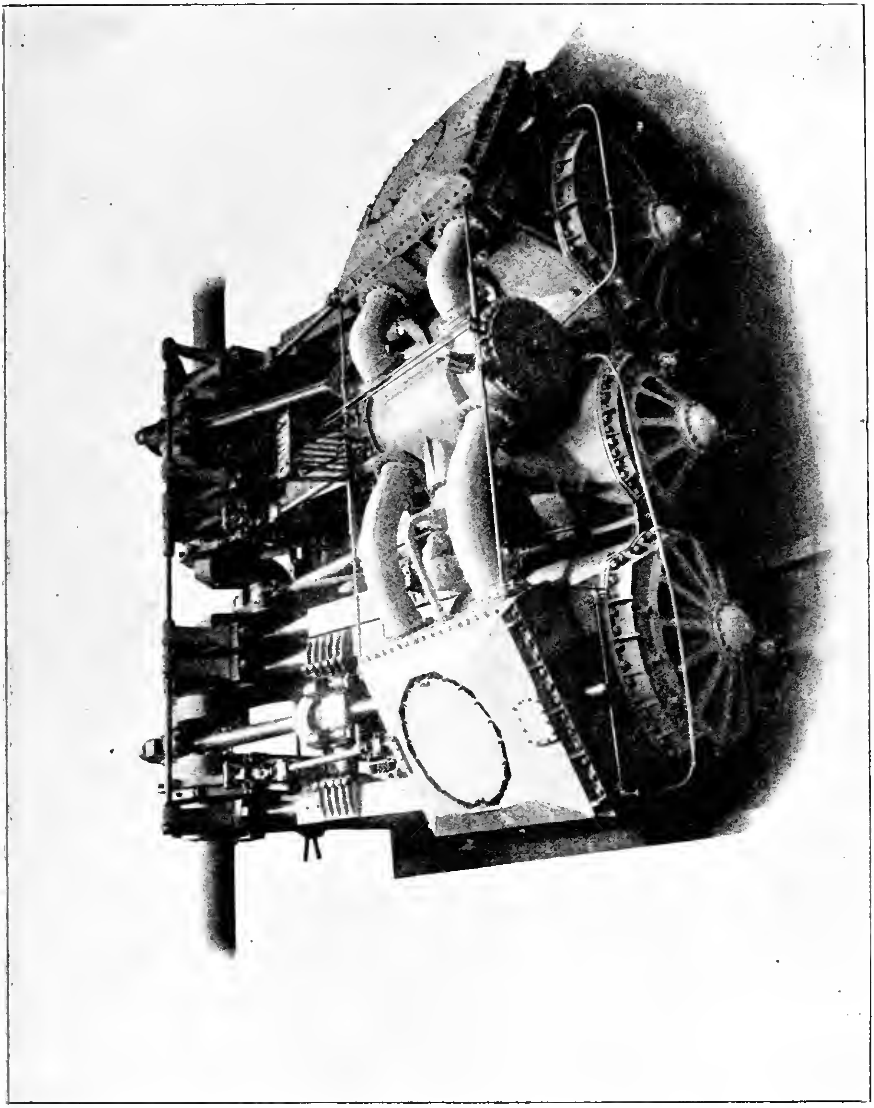


Fig. 21.—Three crank Compound Diagonal Paddle Engines. ("Empress Queen," 10,500 I.H.P., Fairfield Co., Govan.)

with the long stroke of piston which can be permitted with it, allows of a very high rate of piston speed, amounting in some large steamers to as much as 750 feet per minute with a low-pressure piston 108 inches diameter, the

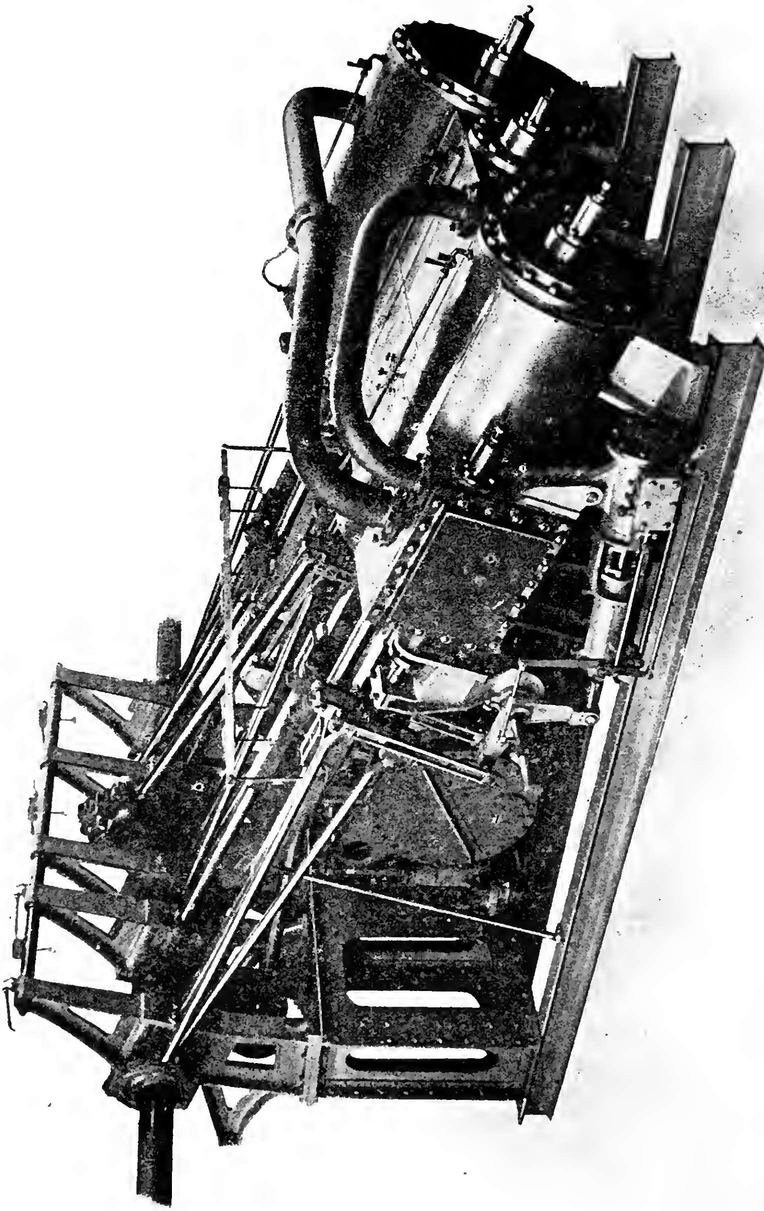


Fig. 22.—Triple Compound Paddle Engine (Campbell & Calderwood)

revolutions in this case being 52; this and other examples of these engines may be studied on Table xiii.

On the American rivers the stern-wheel steamers of shallow draught have the horizontal engine in general use; a remarkable example of it is found in the tugboat "Sprague," of 2,200 tons displacement; her two engines have cylinders 28 inches and 63 inches diameter, and a stroke of piston of 12 feet driving a wheel 40 feet diameter and 40 feet long. At 16 revolutions the I.H.P. is about 2,500.

Beam Engines, by their family likeness to those on shore, show their descent from them; and the survival of type locally is illustrated in their

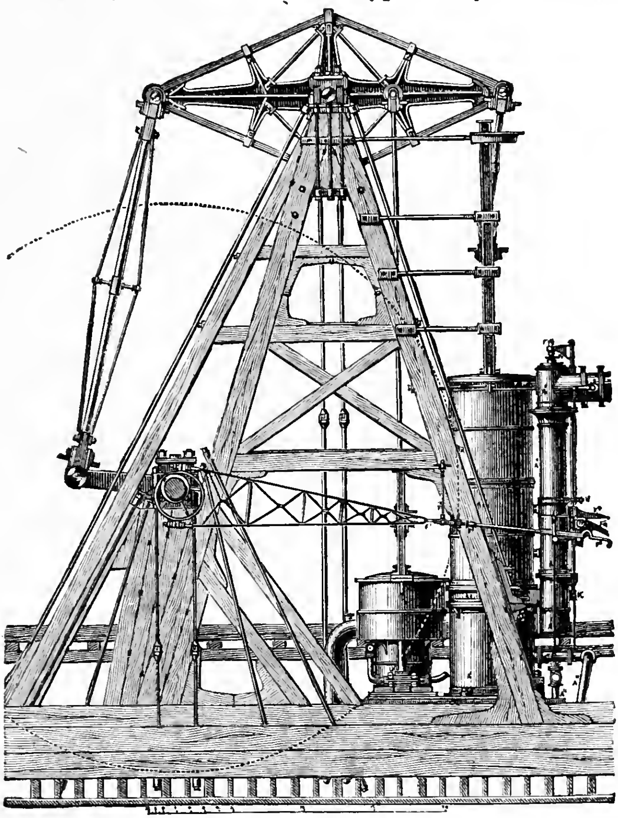


Fig. 23.—American Steamer Beam Engine.

case as it is in that of the direct-acting variety. The second practicable steamboat constructed was Fulton's P.S. "Claremont," built by him in 1807, and used for service on the River Hudson; she was fitted with an ordinary overhead beam engine supplied by Boulton & Watt, from their Soho Foundry at Birmingham, England. Probably, as a consequence, this type of overhead beam engine soon came into general use in America, and continued to be the favourite one till quite modern times, and may be still seen on service to-day with cylinders of enormous size with "drop" valves and wooden framework, etc. (fig. 23). Indeed, it is to this latter fact that it continued so long to hold its own, for wooden structures and wooden connecting-rods properly braced and bolted were permissible with such engines, and were, and are, the peculiar product of the people inhabiting a new country where wood is more plentiful and more easily manipulated than iron. Nevertheless, the machinery of a large Fall River steamer constructed on this method was a splendid example of engineering genius, displaying as it did, the adaptation of what was found to hand to the needs of the problems.

Side Lever Engines are a form of beam engine, and was the type developed in this country from the beam idea. The third practicable steamboat was the P.S. "Comet," already alluded to and shown in fig. 4. This ship was

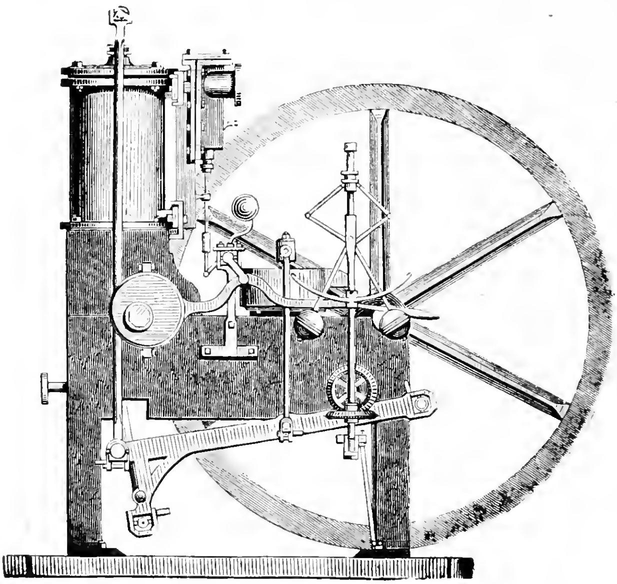


Fig. 24.—Engine of the "Comet," 1811-12.

installed by Bell, her owner, with a single-cylinder engine having a beam at each side, as shown in fig. 24, connected to the piston-rod cross-head and the-crank pin by rods. A modification of this design was adopted many years ago by the builders of tug boats, especially by the Tyneside builders, and called the Grasshopper engine. Such engines are still in use on tugs, and are exceedingly suitable for them. They are very cheap in construction, have a very long stroke of piston for such shallow ships, the racking action when in motion is consequently comparatively slight, and is taken by the keelsons, the stiffest part of the hull; when only one cylinder is employed there is in practice no "dead" point—that is, the crank can be moved by the pressure on the piston from any position in which it may have stopped. This latter quality is due to the position of the connecting-rod with respect

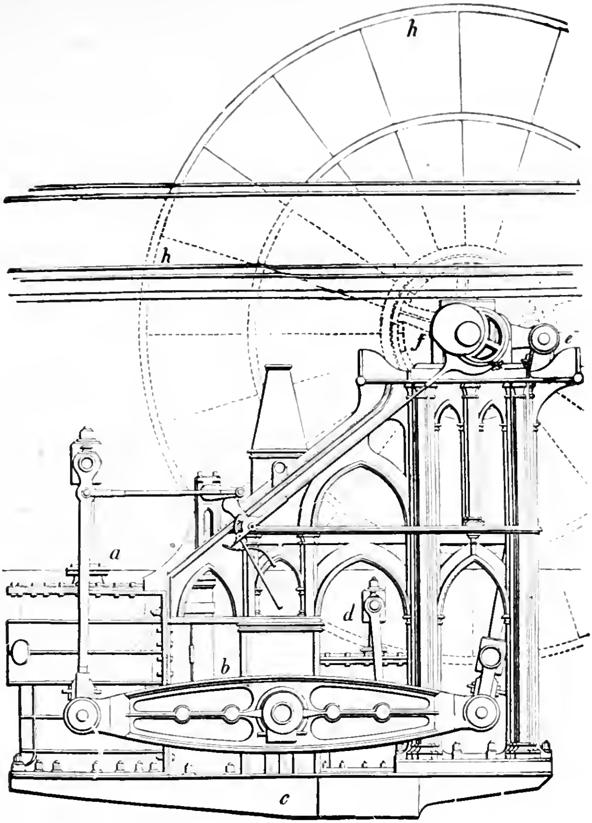


Fig. 25.—" Side Lever " Engine.

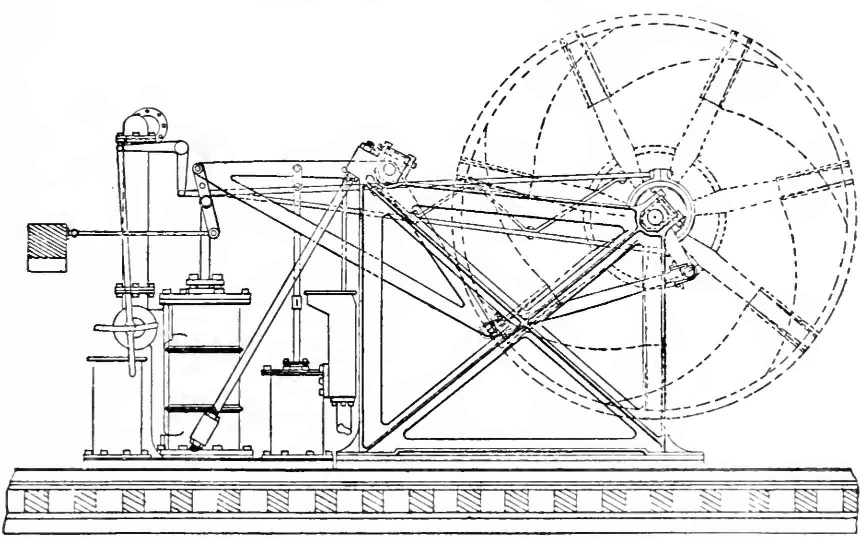


Fig. 26.—Engines of Thames Steamer " Regent," 1816 (Maudslay).

to the levers when the piston is at the end of its stroke, and aided by the slight amount of "play" in the brasses. This class of engine is capable of working satisfactorily when in such a state of disrepair that would in any other form of engine prove dangerous; it is also not exacting in the matter of attention from the attendants.

The most important development of the side lever engine was, however, to

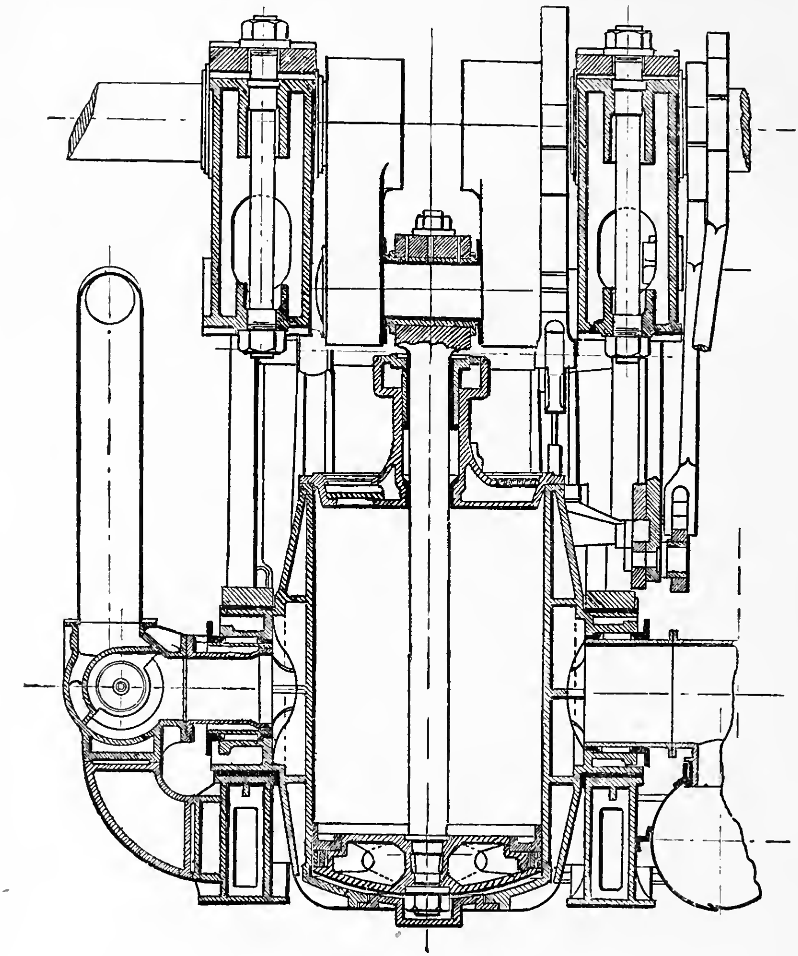


Fig. 27.—Oscillating Engine—Section through Trunnions.

be found in the sea-going steamers of the early half of last century, where it might have been seen in large varieties both of size and design. It was the means whereby the Cunard and some other important companies established the reputations of their ships by regular passages and freedom from breaks down. Fig. 25 is a good example of such an engine, and shows all its salient features. It continued to be the most popular type to the end for sea-going ships, and the "Scotia," the last of the Cunard paddle ships, had such engines with two cylinders, each 100 inches diameter and 12 feet stroke, and developing 4,950 I.H.P. Fig. 26 shows a modified form, inasmuch as the lever is of the *bell crank* type.

Oscillating Engines were first suggested by Trevithick; they were used originally by Dr. Goldsworthy Gurney about 1827 for driving his steam motor cars. In the same year Joseph Maudslay patented the use of such

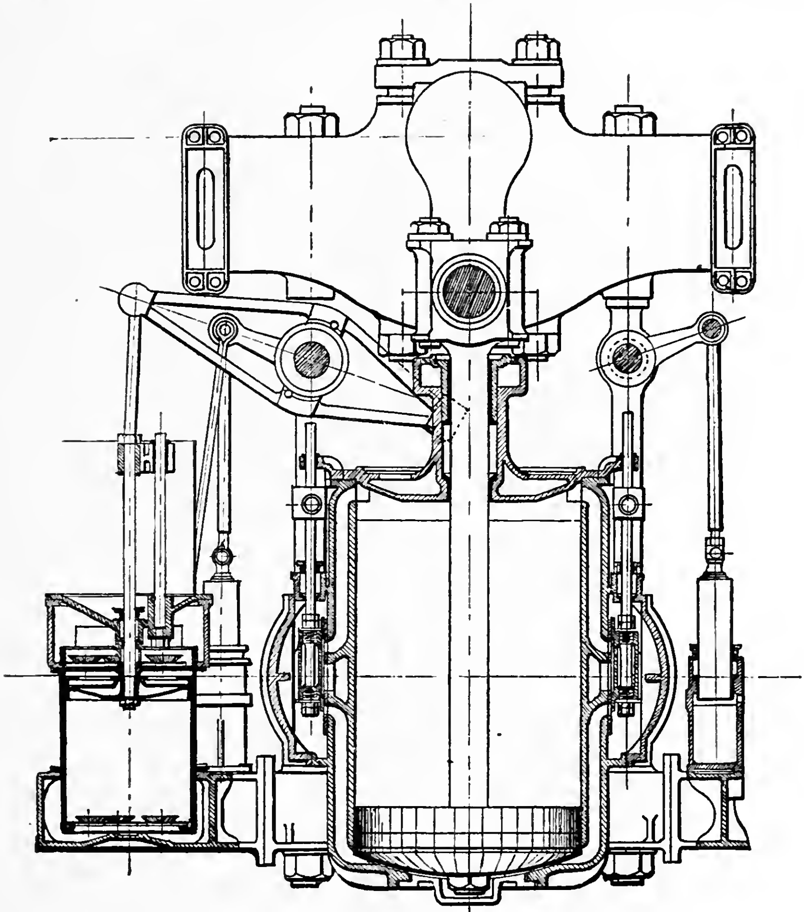


Fig. 27*a*.—Oscillating Engine—Section through Valve-Boxes.

cylinders, and in 1830 William Church took out a patent for their application to driving paddle-wheel shafts. The firm of Maudslay, Sons & Field, however, developed the oscillating marine engine, and until they quarrelled with the Admiralty over some repairs to one, had the sole supply of them. The patronage of the Admiralty was transferred to John Penn & Son for their further supply, and since then this type has been always associated with this firm. It was, moreover, brought by them to a very high state of efficiency, and employed largely on river and cross-channel services, as well as in all kinds of ships in the Navy.

"The Great Eastern" steamship had four oscillating cylinders, 84 inches diameter and 14·5 feet stroke; but perhaps the largest and most interesting example was that of the P.S. "Ireland," 1,952 tons, built by Messrs. Laird for the City of Dublin S.P. Company, for service between Holyhead and Kingston, which, with two cylinders 102 inches diameter and 8·5 feet stroke, 7,000 I.H.P. was developed, and a speed of 23 knots was attained.

The oscillating engine has survived to the present time, as they may still be found in common use on dockyard tugs, and on river craft throughout the world; and although there have been compound engines of this kind with cylinders of considerable size for steam pressure of 60 lbs., the type is not so good for the higher pressures or for triple engines as the diagonal. Figs. 27 and 27*a* show them to be simple in design, having only one rod from piston to crank-pin, requiring no guides, and generally are free from complications, except that the valve-gearing may be considered somewhat complex. They are very light; occupy little space, and permit of a fairly long stroke, even in somewhat shallow ships. In the case of the "Great Eastern," they were placed diagonally two to each crank-pin, and this design has been followed down to modern times for the British Dockyard Tugboats, where each wheel can be worked independently by a pair of cylinders.

It is not necessary to dwell on the other types of paddle engine, as they have had their day and disappeared, except to say that the *Steeple Engine*, as shown in fig. 28, was a favourite one with some Clyde engineers, and it was the type adopted by Messrs. Laird when refitting the P.S. "Violet" with triple-compound engines of 4,070 I.H.P., in which the L.P. cylinder was 108 inches diameter and 6·5 feet stroke. The space occupied by this form of engine was small, and generally permitted of a fairly long stroke, but the thrust of the connecting-rod caused a tilting action on the engine framing, which, unless well provided for by bracing, severely strained the entablature, etc.

Vertical Direct-acting Engines were used by some makers, and with the old sea-going ships with their good depth of hold, and a radial wheel of large diameter, a fairly long stroke could be obtained; but it was always somewhat short compared with that of other types of engine.

Twin-cylinder Engine of Maudslay & Field (fig. 29) is now chiefly interesting as being the one fitted in the first screw ship of the Navy, H.M.S. "Rattler." It was geared to the screw shaft, so that the latter made four revolutions to one of the engine, whose cylinders, four in number, were 40 inches diameter, and the piston stroke 4 feet.

Screw Engines were at first, as in the case of H.M.S. "Rattler," of the same type as used for paddle-wheels (*v.* fig. 29), with spur and pinion gearing connecting them to the screw shaft, and what, perhaps, was of more importance, the necessary revolutions for the screw were got thereby without submitting the engines to such speeds as they were then not fit to bear. Gearing was eventually discarded, and direct-driving adopted, both for safety and economy. The higher speed of engine permitted of the use of smaller ones, which were lighter and cheaper; and as tooth-gearing involved considerable friction as well as frequent repairs, further economies were affected by abolishing it. To-day, however, we are witnessing a revival of wheel-gearing, to permit of the use of turbines in cargo steamers where a small high revolution screw is not permissible, and in ships generally when turbines of high revolution and consequent economy may be employed. It is, of course, under

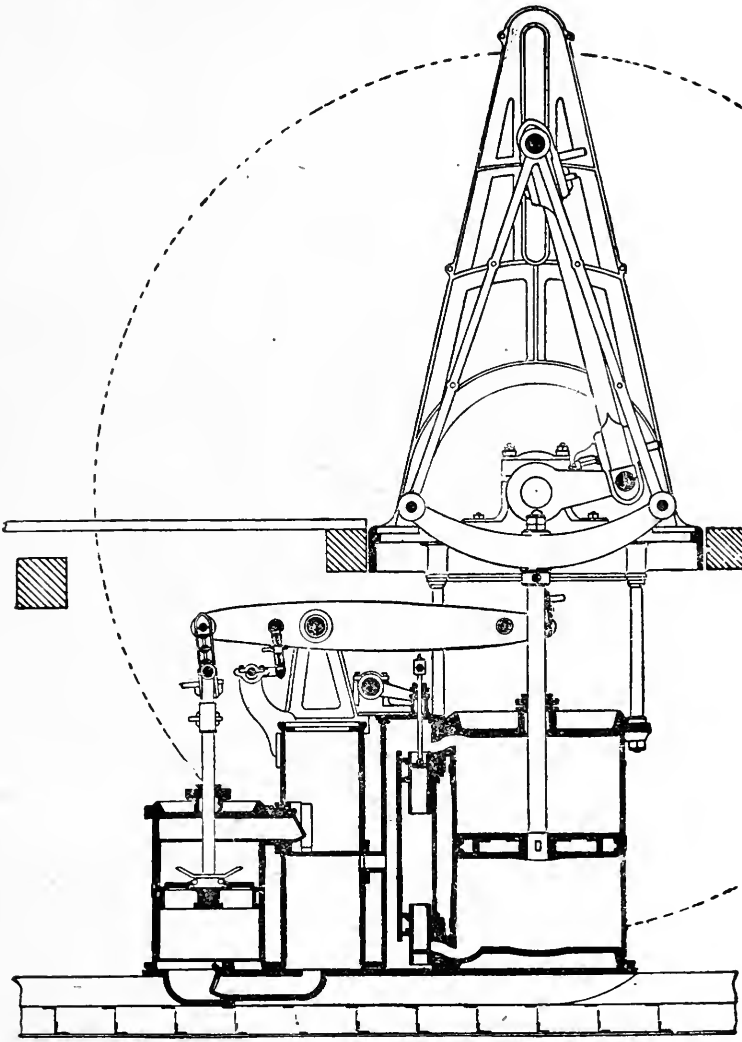


Fig. 28.—Steeple Engine.

much more favourable circumstances that this experiment is being carried out, for in this case the torque of the engine shaft is absolutely uniform, instead of being highly variable ; the pinion is driving instead of being driven ; and, moreover, there are two, one on each side of the spur-wheel ; finally, the teeth of the cog-wheels are to-day double helical machined to exact shape and size, instead of being common toothed of rough cast metal or of wood that, while tough, was not particularly hard (*v.* fig. 45). The efficiency of the helical wheel connection is 98·5 per cent., which is very high, and so far satisfactory, as to permit of high-speed reciprocators, as well as turbines for the drivers.

The Horizontal Direct-acting Engine was the product of Humphrys & Tennant latterly, and in an older form of Boulton & Watt for ships of the Navy; they and one or two other engine builders adopted this type also for the mercantile marine. Although itself now superseded, it displaced and survived the other kinds of horizontal screw engine once so popular on shipboard, and itself only succumbed when it was found that protection against shot could be afforded to a vertical engine in a warship without inconvenience or serious cost. Till then some horizontal type, which was well below the water-line, was considered absolutely necessary, and as the direct-acting kind of engine suited the higher steam pressures of modern time better than the others, it prevailed; moreover, with twin screws a longer stroke of piston was permissible than was the case when only half the breadth of the ship was available for the cylinder and rods. Very early on Boulton & Watt made a horizontal four-cylinder oscillating engine that was fairly successful, and admitted of a longer stroke than the ordinary kind. Experience with the "Simoom" does not seem, however, to have inspired sufficient confidence in them to cause many repeats.

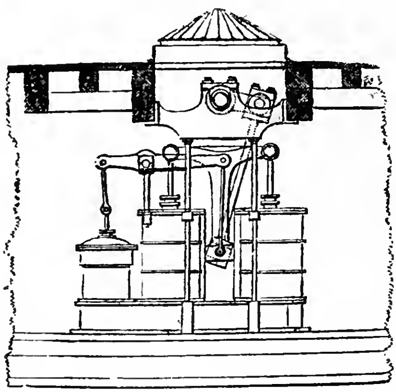


Fig. 29.—Twin-Cylinder Engine.

Such direct-acting engines possess the same good features as the direct-acting paddle engine had when placed horizontally or nearly so; they likewise have the same objectionable features in the way of heavy pistons rubbing on cylinder sides, effect of momentum of the moving parts to cause stresses on the hull, and general vibration notwithstanding balance weights on the cranks; and, finally, the general want of accessibility to some parts when running, and to others, such as the pistons, etc., when requiring overhaul.

Trunk Engines, invented and introduced by John Penn & Son into the Navy in 1849 on board H.M.S. "Arrogant," are of the horizontal type and direct-acting, inasmuch as the rod connects the piston direct to the crank pin, as is the case in the oscillating engine only. Fig. 31 shows such an engine and how the connecting-rod is enclosed in a hollow cylinder or trunk attached to the piston, and with a similar trunk in rear whereby the rod end and gudgeon could be fitted and examined; it also served to ventilate these important parts, and prevent them from overheating. Both in the mercantile marine and Navy they were, in their day, favourite engines; they were splendidly made and finished, and worked even at comparatively high revolutions, and latterly with steam of 60 lbs. pressure quite satisfactorily.

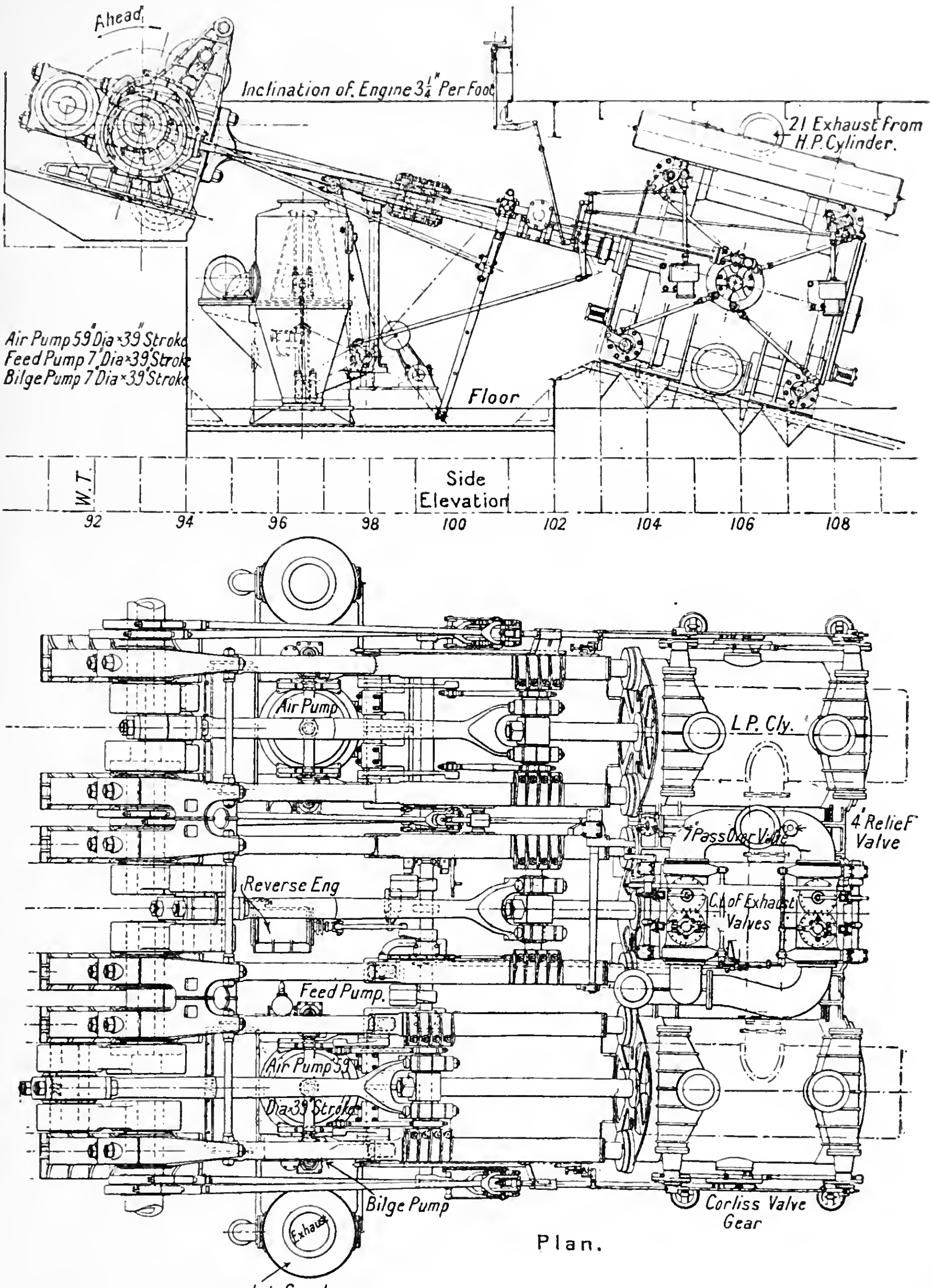


Fig. 30.—Engines of P.S. "C. D.," 6,370 tons. Lake Service, U.S.A. Cylinders 92″-62″-92″ diam. × 102″ stroke. 7,600 I.H.P. at 28 revs.

There was, however, considerable heat loss by radiation from the trunks, and condensation in the cylinders by their cooling action; the friction at the stuffing-boxes was also somewhat of a drawback, being often considerable, and could be easily made so great as to slow down the engines.

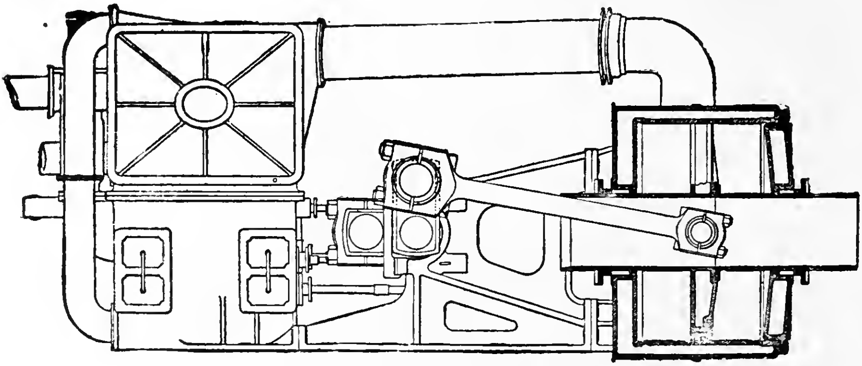


Fig. 31.—Trunk Engines.

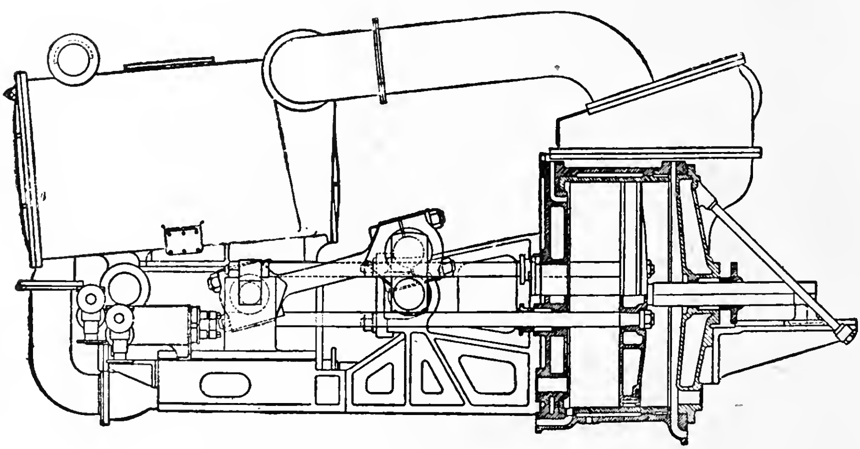


Fig. 31*a*.—Return Connecting-Rod Engines.

Their screw was generally "left-handed," so that the thrust of the connecting-rod when going ahead was upward; but when at full speed it was considerably in excess of the weight of the piston, so that the rubbing was sufficient to "barrel" the upper side, and when in stern gear the thrust and weight were combined in doing the same thing to the lower side

The Return Connecting-rod Engine is shown in fig. 31*a*, by which it will be seen that it is a steeple engine laid horizontally with the two piston-rods prolonged to the gudgeon cross-head instead of into a banjo-frame. This

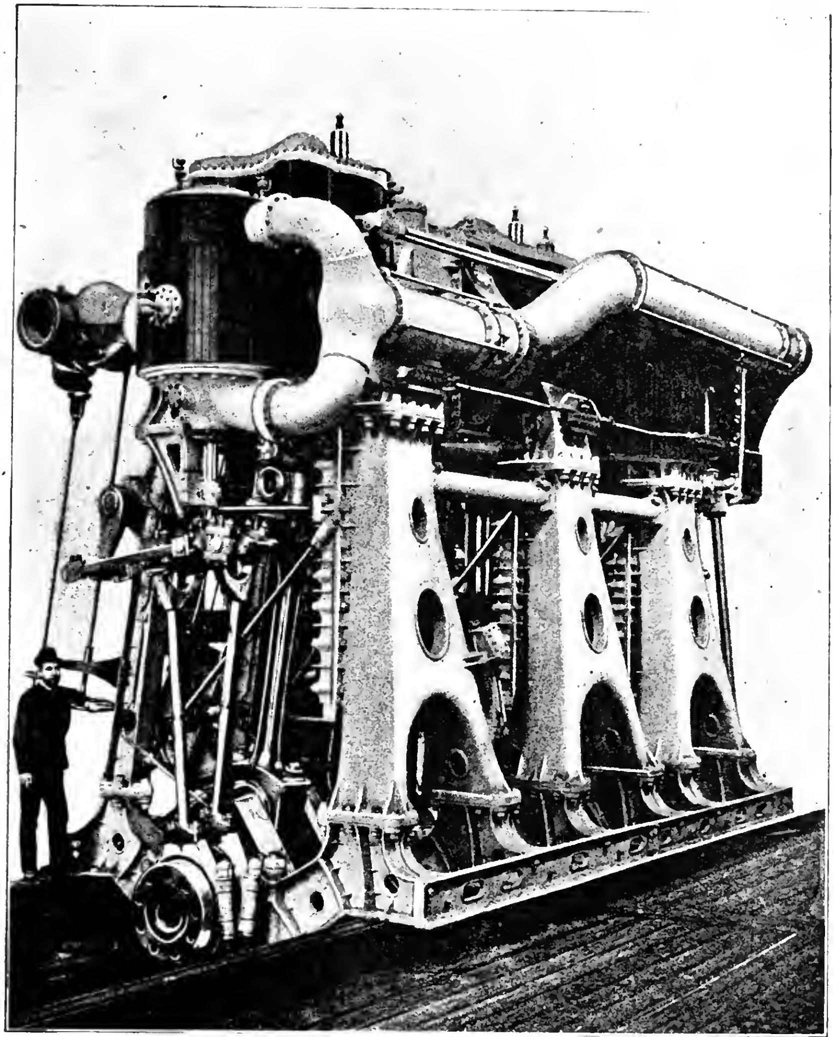


Fig. 32.—Three-Crank Triple-Expansion Engine (Naval).

type was introduced by Maudslay & Field to the Navy, and used by them till the vertical engine displaced them. Several other builders also adopted this type, inasmuch as thereby a much longer stroke was possible with ample room behind the cylinders for examination and repair, and all the working parts were quite accessible when moving. An attempt was made latterly

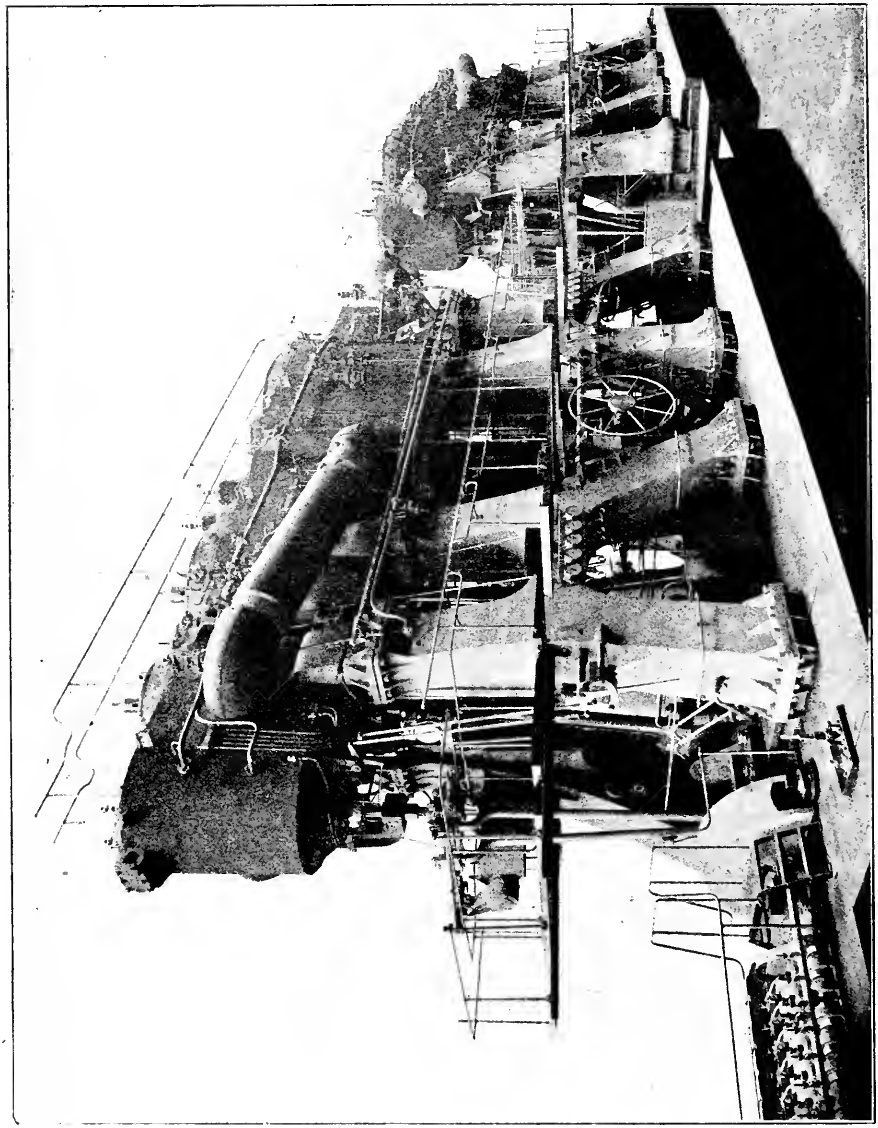


Fig. 33.—Vertical Quadruple Expansion Engines—T.S.S. "Kinfauns Castle" and "Kildonan Castle," 10,000 I.H.P., Fairfield Co., Govan.

Cylinders 28, $39\frac{3}{4}$, $57\frac{1}{2}$, and 82 inches in diameter × 60 inches stroke.

to support the pistons by fitting a substantial tail rod with a slipper support and guide at the back; this arrangement was only a palliation, and did not prevent rubbing of the cylinder walls at bottom. With the vertical engine there is no absolute need of such a fitment, and although the tail rod through their cylinder covers tends, no doubt, to steady the piston, and when the ship is rolling to prevent excessive side play.

The Vertical Direct-acting Engines.—Figs. 32 and 33 are now the universal type of screw engine employed throughout the world for both war and merchant ships; its advantage over others are so obvious as to need no further comment, after having pointed out the defects of those others whose competition lasted. This type was first employed by the Thomsons, who founded the business now so extensive and successfully carried on by John Brown & Co. at Clydebank. In the mercantile marine it quickly found favour, and was soon adopted by most builders of marine engines for it, and is still largely used for generating electricity on shore. For naval purposes it was deemed unsafe formerly, as having the vital parts of the machinery above the waterline, therefore much exposed to shot and shell in a way not obtaining with the horizontal engine. This objection was, of course, from the military point of view, and formed a bar to their use long after the introduction of armour plates had provided a means for their protection if naval designers had so desired, and as was eventually adopted, and still prevails.

In the mercantile marine, not only is the foundation or bed-plate of cast iron, but the columns supporting the cylinders, both front and back, and forming guides for the piston-rod ends, are often of cast iron. In the Navy, for lightness and strength, and in a measure to resist concussion, the columns are of forged or cast steel, and the foundations themselves also of steel. Figs. 32 and 33 show two different makes of engines, the one for the Navy, the other for an express steamer. Fig. 32 is that of an engine with cast-steel foundation and front column with cast-iron back columns. In fig. 33 the columns on both sides are of cast iron, as also are the foundations. In fig. 34 is seen an example of a wrought-steel structure of extreme lightness, and yet so rigid and strong as to permit of 380 revolutions per minute being run quite satisfactorily. The cost of the latter, and such structures as seen in fig. 35, compared with either of the others, but more especially with the engines of fig. 33, is very high, but, notwithstanding this, the saving in weight, combined with the perfect rigidity, warrants the use of similar structures for the support of the cylinders of express steamers of quite large size. Some builders of mercantile engines prefer these wrought-iron or steel columns for the front of the engines of all kinds (*v.* fig. 36), as they permit of freer access to the moving parts; moreover, when secured in the foundation by a single nut and to the cylinders in the same way, so that these columns can be turned down from a plain rolled bar, they are even cheaper than cast-iron ones.

In comparing the engine design and practice of to-day with that of past generations, it must not be forgotten that the earliest engineers had to make almost every part of cast iron; for there were no steam hammers, and the forgings made by the tilt hammer and the rolled-iron bars of that period were of very limited size and form. Steel could be had only in very small pieces, and was very dear. Modern engineers have an unlimited supply of splendid mild steel, and now even reliable high tensile steel can be obtained at quite moderate costs. Forgings, rolled bars, and plates of huge size can be obtained cheaply from various sources of supply, and quickly compared with the time required twenty-five years ago. Our predecessors knew not the advantages of "dumping," or of steel castings, nor did they enjoy such white metal and strong zinc bronzes as we know them.

Arrangement of Cylinders in the modern marine engine is a much more important problem than was the case when there were two cylinders only.

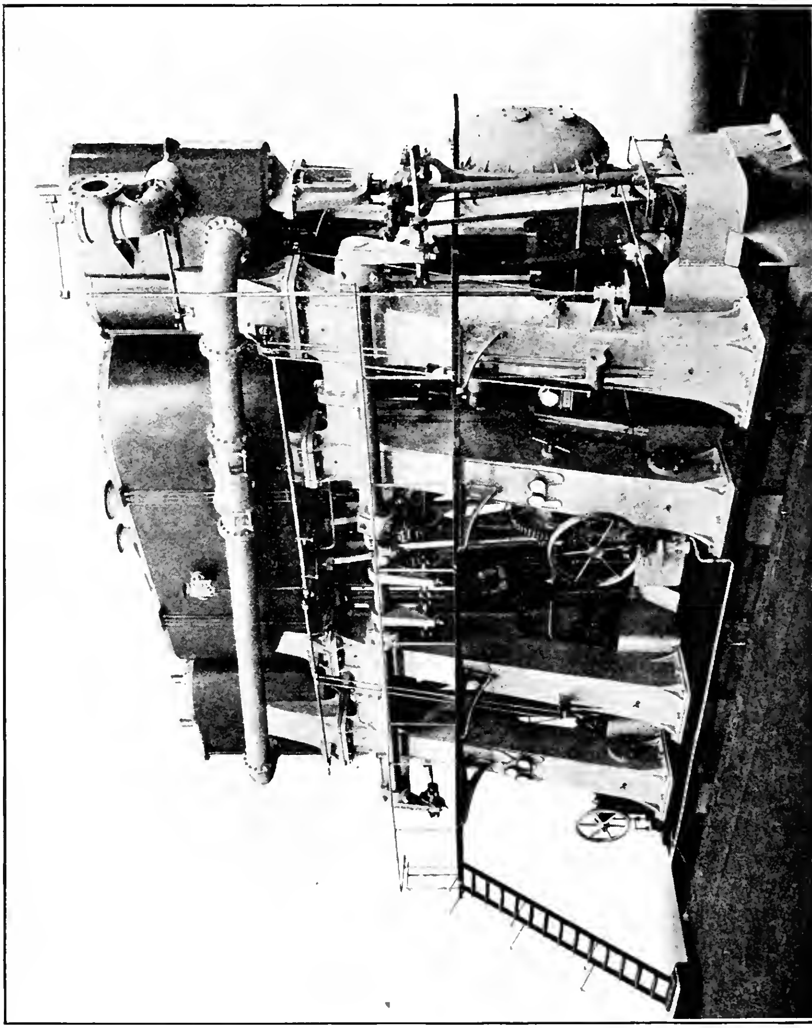


Fig. 33*a*.—Mercantile Quadruple Engines.
Cylinders $26\frac{1}{4}$, $37\frac{1}{2}$, 59, 79 inches × 57 inches stroke; pressure 230 lbs. (By Richardson, Westgarth & Co., Ltd.)

To-day, with a multiplicity of cylinders, their arrangement is governed by circumstances not obtaining and never dreamed of formerly. We have now to consider the question of balancing the momentum of the moving parts, as well as to determine what is the best sequence for the flow of steam through the system from the boilers to the condenser to attain the greatest steam efficiency, together with the minimum amount of piping for that purpose. These and other practical considerations impose themselves on

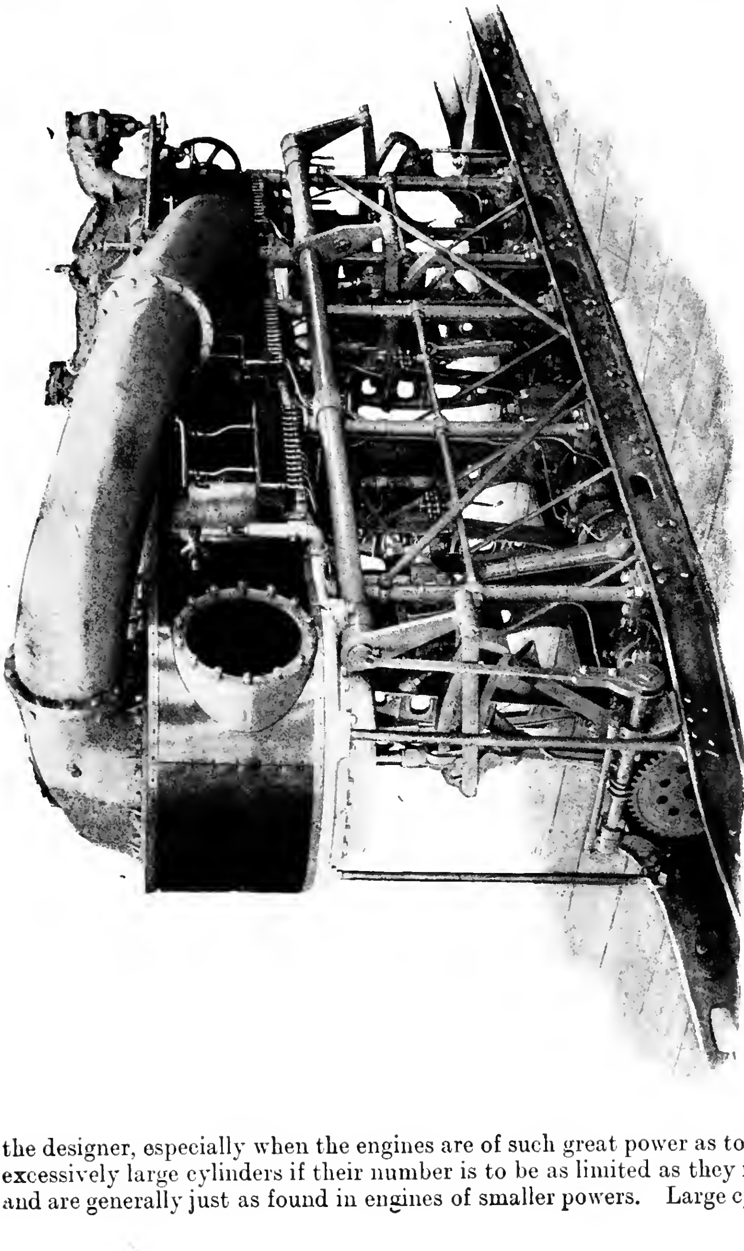


Fig. 34.—Engines of H.M.S. "Salmon" (Destroyer), 4,000 I.H.P. Designed by the Author.

the designer, especially when the engines are of such great power as to require excessively large cylinders if their number is to be as limited as they may be, and are generally just as found in engines of smaller powers. Large cylinders can be, of course, manufactured, and large pistons, being now of solid cast steel, are no longer so heavy and cumbersome as to be almost impossible

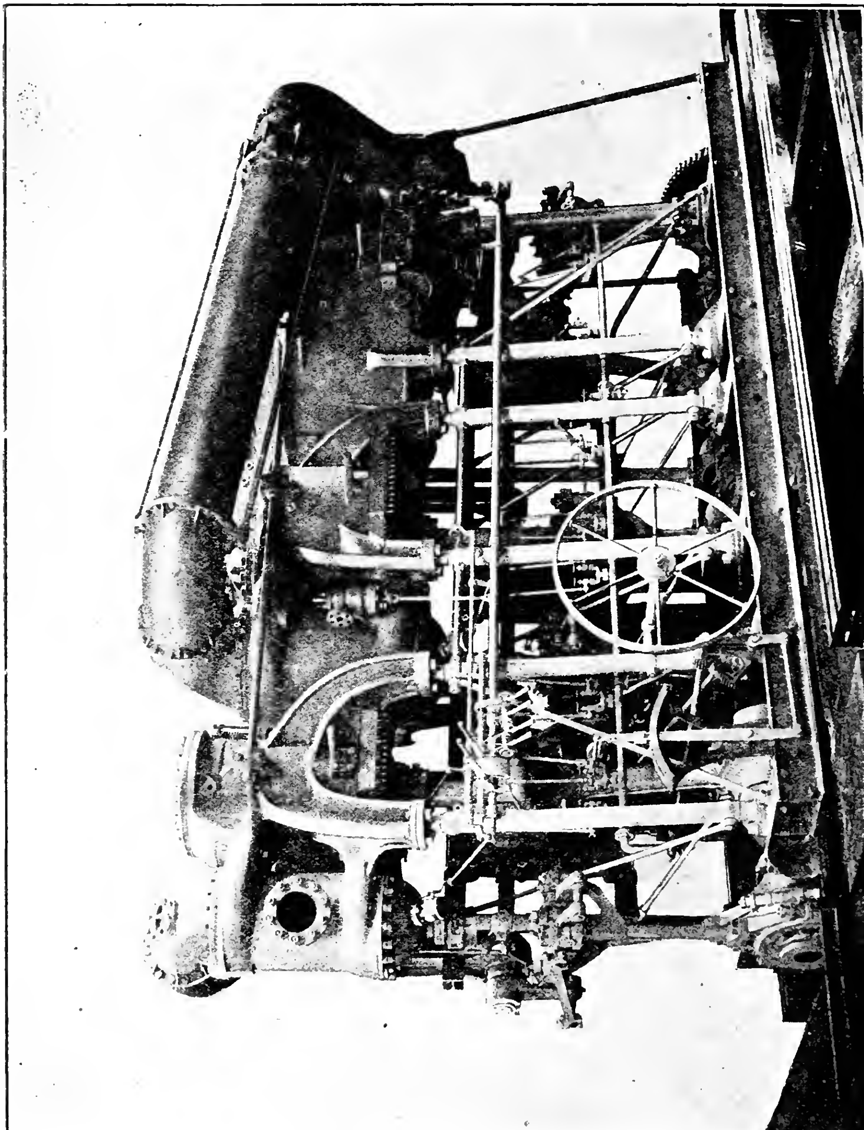


Fig. 35.—High-Speed Cruiser Engines. 7,500 I.H.P., Combined.
Cylinders 20½, 33, 54 inches diameter, 27 inches stroke, 220 revolutions.

of handling with the engine-room appliances. Moreover, all engines to-day are of the vertical type; there is, consequently, no longer any fear of damage

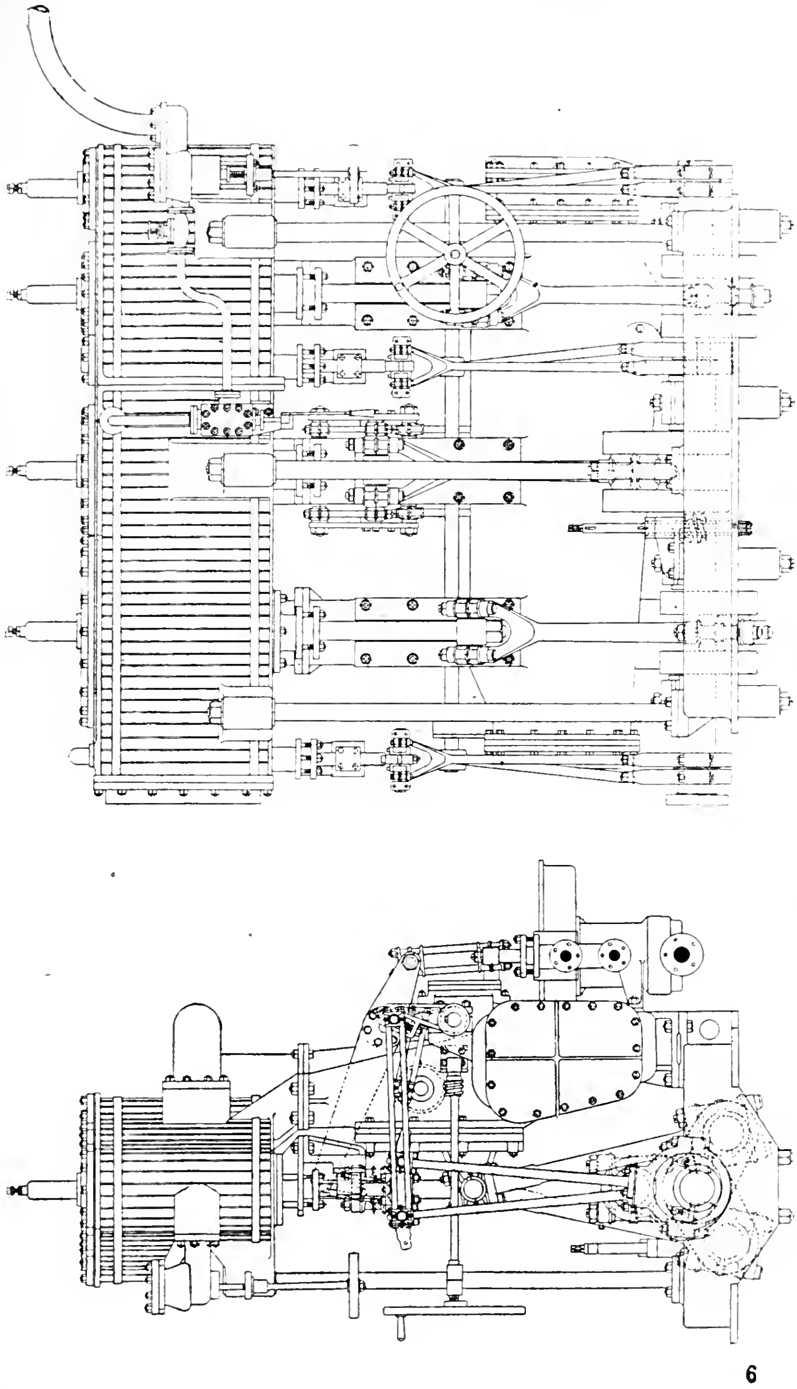


Fig. 36.—Triple Compound Engines of 400 I.H.P.

Cylinders 12, 20, 32 inches diameter; stroke 23 inches; pressure 180 lbs. (As designed by Mr. A. H. Tyacke for fishing and similar craft.)

to cylinders, liners, and walls, or difficulties due to the position of the cylinders whereby the use of mechanical means for getting them in and out of place was precluded, besides rendering them difficult to examine. But the risk of casting large cylinders is still great; the cost of replacing such an one, if damaged, great, and perhaps, what is more important in these days of high piston speed, there is always the impossibility of designing the low-pressure ones with ports of sufficient size for a flow of steam slow enough to be consistent with high efficiency. The area of port possible for a cylinder with slide valves varies practically as the diameter, inasmuch as the length of port axially does not increase rapidly with increase in the diameter. The volume of steam passing through it varies as the square of the diameter. When piston speeds seldom exceeded 500 feet per minute, ports with a sufficient area for very large cylinders could be designed easily enough; but with a speed of piston of 1,000 to 1,200 feet it is impossible to provide them large enough for moderate flows and with passages small enough for moderate "clearance."

With the Diesel and other oil engines, where the initial pressures are very high, while the mean pressures are quite low, the latter condition requires a large total cylinder capacity and the former small units; hence such engines have as many as twelve cylinders per engine when of the large horse-power required on shipboard for good speed of ship.

Even in the steam engine of to-day the boiler pressure, and consequently the initial pressure in the H.P. cylinder at starting is as much as 230 lbs., while the mean pressure of the compound cylinder system is not very much more than that obtaining when it was a fifth. In the Navy, with water-tube boilers, the pressure is often even greater still, but the referred mean pressure in such ships is much higher than in the mercantile marine when running full speed.

With a Compound Engine the least number of cylinders is of course two, which may be in line axially with their pistons attached to a common rod, and operating on one crank by one connecting-rod (*v.* fig. 75). Such engines were at one time used in certain cargo steamers, and thought well of at the time, in large measure owing to the small space occupied by them and their comparative cheapness. More commonly the two cylinder compound engine had them side by side, each operating on a separate crank, the one opposite the other when the steam expanded direct from the H.P. to the L.P. cylinder, or set at right angles with a receiver between the cylinders into which the H.P. exhausted, and from which the L.P. took steam. In some few cases the cranks were set at a different angle, generally about 105°, so as to favour the timing of exhaust from H.P. and flow to the L.P., and to reduce thereby the variation in pressure in the receiver. These latter engines were, however, somewhat unhandy in starting and reversing, and eventually were given up in favour of the cranks at 90°, and having a somewhat larger receiver between the cylinders.

The Three-cylinder Compound Engine, each cylinder with a separate crank and connections, was deservedly a favourite one for large powers; for not only was there with it the advantage of having the L.P. cylinders of moderate size, but a much higher initial pressure could be thereby maintained in them, consequently with the minimum amount of drop between H.P. and L.P. cylinders; a fair division of the work between the three cranks was made, and they were usually set at angles of 120°. This engine was a favourite for electric generation on shore stations with large units.

Four-cylinder Compound Engines enabled the same subdivision of the L.P. pressure member of the system without resorting to an additional crank by dividing the H.P. member, and placing one H.P. cylinder tandem with one L.P., each pair side by side operating its own crank, which was at 90° with the other. This type of engine was adopted by Messrs. Maudslay, Sons & Field in the celebrated White Star steamers, "Britannic," etc.; moreover, it was the method used by many engineers as a cheap and ready way of compounding the old expansive engines after the superiority of the compound system was assured about 1870.

Six-cylinder Compounds, each pair of H.P. and L.P. being tandem and side by side, each operating on a separate crank, as with the four-cylinder system, of which it is an extension; this design was adopted for the "City of Rome," as a further subdivision of cylinder due to her then large power.

With a Triple-expansion Engine the least number of cylinders is, of course, three, as with a quadruple expansion it is four. It does not, however, follow that each must have its own separate crank, as is the rule with the single-acting oil engine; and, although it is common practice now to do so with the marine steam engines, it was a few years ago not an unusual thing to find a triple-expansion engine with two cranks and the cylinders as in fig. 73, Nos. 1, 2, and 3. Moreover, the quadruple engine when first placed on the market as a competitor with the triple had only two cranks (fig. 73, No. 1); this, as a matter of fact, was at that time placed to its credit as a means whereby it occupied no more space than a common two-cylinder compound, and less than a three-crank triple or compound engine. When, however, the demand for a non-vibrating engine was insisted on by the Admiralty, and much desired by all interested in the passenger service, the Yarrow-Schlick-Tweedy system of balancing the four-crank quadruple engine gave it such an advantage over the three-crank triple that makers of the triple engine had to adopt the four-crank arrangement, and with two L.P. cylinders, either as Nos. 3 or 4 in fig. 73, to obtain like advantages.

A Single-crank Engine is seldom or never seen to-day, except in quite small and cheap launches, or harbour service boats. Even single-crank tandem compound engines, which at one time were much in favour with one or two shipowners and some few engineers, are no longer made. Single-crank paddle engines are also a thing of the past.

Two-crank Engines still survive in paddle steamers of moderate power with compound cylinders (*v.* fig. 19); with the larger power compound engines three cranks are favoured (*v.* fig. 21), as each of the two L.P. cylinders are of moderate size, and the ratio of maximum to mean torque is lower than with two cranks, consequently the movement of the wheels is less jerky, and a smaller shaft possible. Triple-expansion paddle engines are invariably of the three-crank type, as shown in fig. 22.

Two-crank Screw Engines are still made with compound cylinders for quite small powers, as in tug boats, steam launches, and other small craft; they may be adopted with advantage still in cases where prime cost, weight, and space occupied are of more importance than fuel consumption. With triple-compound cylinders such engines are now very seldom, if ever, made, but with quadruple-compound cylinders something like those shown in fig. 75, the two-crank arrangement is quite a good one for small craft, especially those having water-tube boilers for the sake of quick raising of steam, as in yachts or their launches.

Three-crank Screw Engines (*v.* figs. 32 and 36) still continue to be the common and, to a great extent, the favourite form in the mercantile marine, when the power is not great, and the service either cargo-carrying or combined with a passenger service of not high class, inasmuch as it is cheaper, occupies smaller space than the four-crank, and can be balanced sufficiently

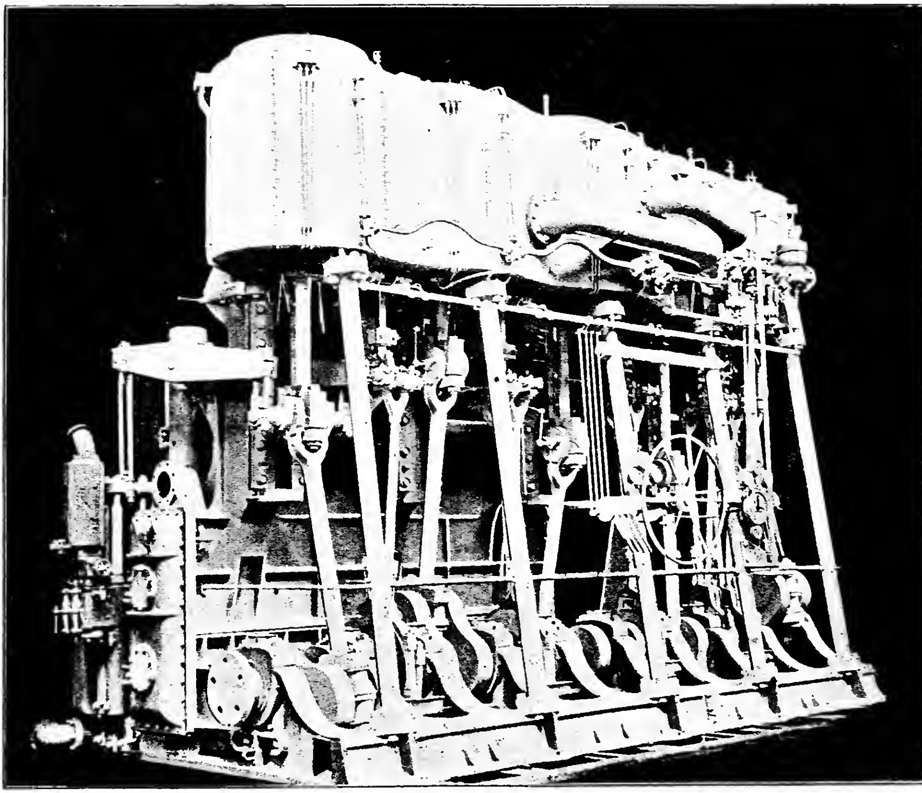


Fig. 37.—Five-Cylinder Quadruple-Expansion Engines of s.s. "Inchdune" (Central Marine Engine Works, West Hartlepool).

well, if thought necessary, by simple and comparatively inexpensive means, so as to cause little or no inconvenience from vibration.

The Four-crank Engine (figs. 33 and 33*a*) is, however, more easy to balance, and the balancing, when done, is more perfect; the quadruple system permits of the use of higher boiler pressures with higher efficiency and, therefore, with advantage; on this triple-expansion system, however, there are two L.P. cylinders, with such advantages as arise from their smaller size.

It may be taken, then, that as a rule for express passenger ships and warships, the four-crank engine is a better one to adopt; for very small ships, or where the engines are small, the three-crank engine has advantages which may outweigh those which would favour the four-crank in a general problem.

Five-crank Engines (fig. 37) have been made to a limited extent with cylinders on the quadruple system, thus having two L.P. cylinders. It has been claimed for them that there is a superiority in the matter of balancing together with a steam consumption as low as any other. Such an engine may be employed with advantage when, owing to size, it is desirable to have two L.P. cylinders; but for small power it seems an unnecessary expense with considerable complication; moreover, the mechanical efficiency cannot be so good as that of a three-crank engine of the same power.

Six-crank Engines for steam have been used to a very limited extent by marine engineers. Fig. 38 is a good example of such an engine of very large power. Oil engines have now come into use on ships of considerable power, so that six cranks are common; they are generally in two pieces, and so coupled that No. 1 and No. 6 cranks are in line, as are also Nos. 2 and 5 and Nos. 3 and 4, each pair being at angle of 120° with the others. With the two-stroke cycle they may be in sequence at 60° angles.

Eight-crank Engines will also be used largely, as they are already for internal combustion systems, when the power is very great, since there is a decided limit to the size of the cylinders of such engines. It may be that steam engines will also be made with these numerous cranks to compete with such engines and turbines.

Of Oil Engines there are three kinds used on shipboard—viz., the *petrol*, the *paraffin*, and the *crude* or *heavy oil* engine.

(1) **The Petrol Engine**, requiring as the fuel supply the light volatile oil of that name, having a flash point from 80° to 100° F., is not admissible on board ordinary passenger ships, from the danger attending the carrying and storing such a highly inflammable liquid. It is, however, used extensively on launches and other small craft for coasting work or on rivers and small lakes, and is as efficient and convenient in them as in motor vehicles on shore. The ease of starting a cold engine is always a strong recommendation for this oil.

(2) **The Paraffin Engine** using a refined light oil obtainable almost everywhere, and safer to carry, use, and store than petrol, inasmuch as it is not nearly so volatile, and its flash point is considerably higher—viz., 120° to 150° F.—is employed for many purposes now, as it is almost as easy to start as the former and runs quite as well. Both it and the petrol engine work on the usual Otto or four-stage cycle, and require ignition by an electric spark, which may be produced by a secondary battery or by a small dynamo worked by the engine itself, called a *magneto*.

(3) **The Heavy or Crude Oil Engine**, which is the desideratum for shipboard, on account of its comparative safety, uses oil as fuel whose flash point is over 200° F., and not volatile enough to be used in the same way as the light oil engines. Moreover, as the oil is unrefined, or else refuse, special treatment has to be accorded to it for the different varieties of fuel used. Texas, Batoum, and similar oils practically free from bitumen may be used with comparative ease with suitable carburetters, etc., and with suitable engines and care will run sufficiently well as to be used for driving dynamos for power purposes, and most engines of this kind can be trusted, therefore, to drive propellers of ships quite well for considerable periods if there is no

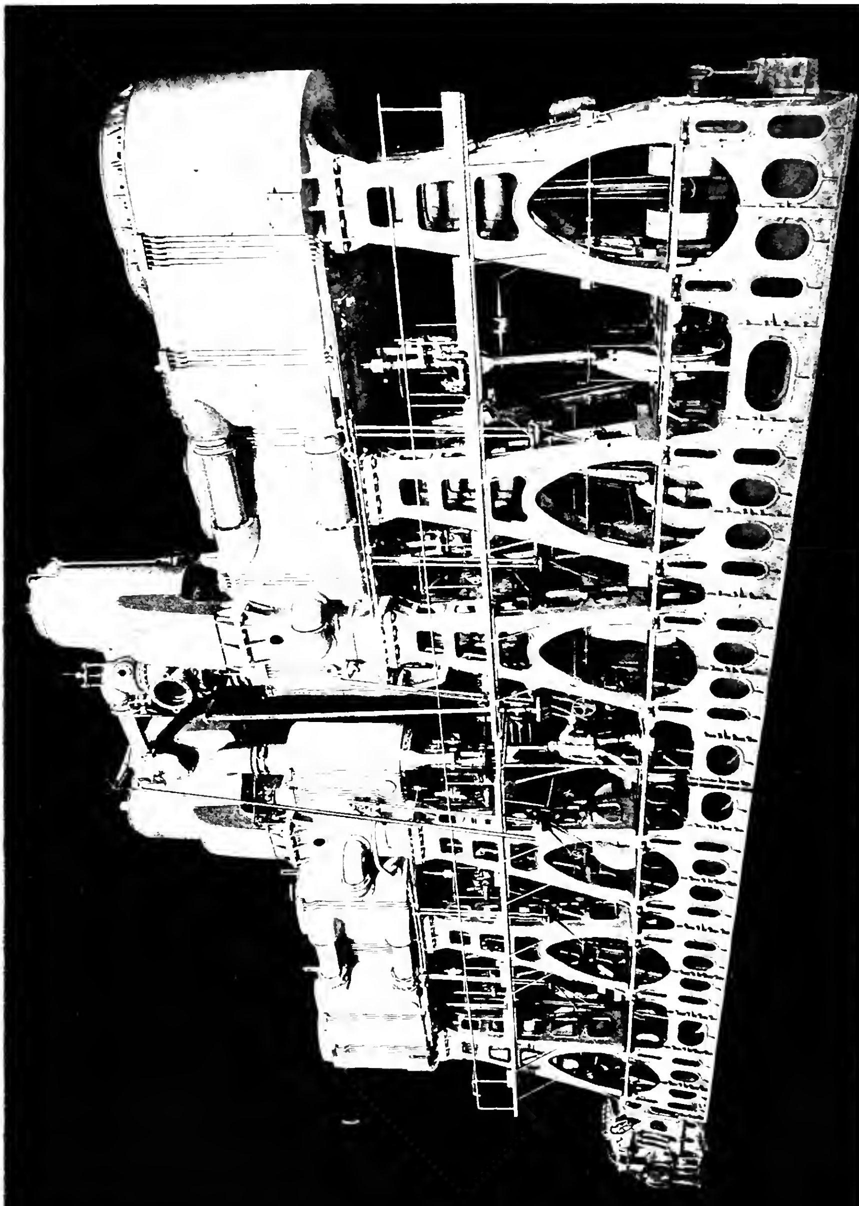


Fig. 38.—Quadruple Eight-Cylinder Engines of T.S.S. "K. W." 40,000 I.H.P.

Cylinders 112·2, 74·8, $\frac{37 \cdot 4}{49 \cdot 2}$, $\frac{37 \cdot 4}{49 \cdot 2}$, 74·8, 112·2 inches diameter × 70·8 inches stroke.

stoppage at any time sufficiently long to permit of the cooling of the cylinders, etc. With a large number of the oils obtainable at other parts of the world the crude oil contains varying amounts of bituminous matter, which not only causes excessive deposits of tarry matter, but what is worse, a formation of coke, hard and refractory, which, if deposited in the cylinders or among the valves, will cause serious damage to them.

The Griffin and some other engines were designed to run with the gas or vapour distilled from crude heavy oils and their residuals by the high temperature of the exhaust gas; this vapour, mixed with air, was drawn into the cylinders as in the ordinary gas engine and exploded there by a spark. The tarry matter and pitch were in this way thrown down in the distiller and excluded from the internal parts of the engines, so that they remained clean and free from grit. The early promise of these engines was unfortunately not maintained as fully as was desirable, and after the Diesel engine had proved so successful it displaced them. The cylinders, etc., were kept sufficiently cool by the usual water-jacketting.

The Diesel Engine,* which is now largely used for ship propulsion, differs fundamentally from other oil engines, inasmuch as it draws in a charge of *air only* and compresses it highly, generally to about 500 lbs. per square inch, when it becomes so hot as to ignite with certainty the spray of oil injected into it at the commencement of the active stroke; moreover, the ignition is gradual instead of instantaneous, and the pressure is practically that at which compression ceased, so that there is no shock due to explosion; expansion commences before combustion is complete, and continues to the end of the stroke, when the products escape through the exhaust valve, being driven out by the piston of the four-stroke cycle and by a blast of fresh air in the two-stroke cycle. Almost any oil will do, but generally heavy crude oils freed from highly volatile constituents, or residuals of oils from which petrol, paraffin, and lubricating products have been extracted, are used; in this country tar oil and shale oil residues are home products which can be used with satisfactory results, their flash point is about 220° F.

The Semi-Diesel Engine, which also uses similar fuel, works with less compression, about 150 lbs. or 60 per cent. of the initial pressure on ignition, and consequently requires a hot plate or bulb on which the spray impinges to effect it. It works on the two-stroke cycle, and is scavenged by air pumped into a receiver by the underside of the piston. It is fairly efficient, and much used on small ships with cylinders up to 16·5 inches diameter.

The Diesel engine, like the other oil engines, was single-acting, working on the four-stage cycle; air only is admitted on the first descent of the piston, on its return it is compressed, sometimes to the extent of 40 atmospheres, so that even with means for cooling the cylinder the temperature is very high. Just as the piston is about to make a second descent the necessary supply of oil is sprayed into the cylinder top by means of a jet of air compressed higher than that in it; the finely pulverised oil at once ignites, and burns during the early part of the stroke, and so maintains the pressure attained by compression (*v.* fig. 101); at the end of the stroke the exhaust valve is opened, and the piston on returning to the top scavenges the cylinder —that is, drives out the products of combustion. It again descends, drawing in air alone as before.

To compete with the steam engine, especially on shipboard where weight and space are of importance, Dr. Diesel, and those acting with him at Nuremburg, devised the double-acting two-stage cycle engine, whereby an explosion takes place at each end at every revolution, so that its activities, so to speak, are equal to those of the steam engine. With the two-stage cycle,

* *Vide* Appendix A

whether the engine be single or double-acting, the *modus operandi* is the same, and as follows (*v.* fig. 51) :—The cylinder is charged with air alone, as before, but now it is forced in under quite moderate pressure by a special pump worked by the engine, much in the same way that the air pump of a steam engine is worked (*v.* fig. 68). The piston compresses, and the oil is injected quite as before, so that when it descends for the first time it is filled with products of combustion. An exhaust port opens at the end of the stroke, and by another the admission of air is made in sufficient abundance to thoroughly scavenge it, and leave it filled with pure air. In the case of the double-acting engine the exhaust port is at the middle of the cylinder, and the piston sufficiently deep to act as the valve for each end to close it before compression, and keep it closed during explosion and expansion. This, of course, means a very long cylinder, and a very hot one, too, when there is an explosion every revolution at each end. Both it, the piston, and piston-rod will require to be water-cooled somehow.

The oil consumption of the Diesel engine when in good working order is very low, under half-a-pound per horse-power developed, and in some cases it has been as low as 0·38. This latter is equal to 0·475 of best Welsh or 0·51 Newcastle coal, or less than half that of a turbine or quadruple-expansion reciprocator, but much more lubricant is expended.

It is estimated that of the total heat of combustion in the cylinder of the Diesel engine 40 per cent. is usefully employed, 40 per cent. passes away with the gases at exhaust, and 20 per cent. is absorbed by the cooling water. The cycle of operations in its cylinders may be followed in fig. 51, Nos. 1, 2, 3, and the engine itself with its various pumps in fig. 50*a*, and in Appendix A.

The Reversal of Oil Engines is accomplished by using one or more of the cylinders as an air engine, supplying it with compressed air carried for the purpose in small craft, or accumulated by the oil engine when running, and stored for the purpose as it is in larger ones (*v.* fig. 50*a*).

Reversal of the Propeller, when driven by oil engines, can be accomplished by means of wheel-gearing, as in a motor car, or, better still, by twisting the blades sufficiently to bring the base or part of them near the boss to a transverse position of no pitch, when the outer part and tips will have reverse pitch sufficient for navigation purposes. This was done in various ways, and one similar to the method adopted by the late Mr. Bevis for modifying the pitch is quite a successful one. In this case, instead of a nut on the shaft, there is a sleeve, which is free to slide on the shaft, and is carried round on it; it is grooved as is a sliding clutch, and the lever operates in the same way as in a clutch gear. In head gear the two blades of the propeller are of true pitch and the usual form to be highly efficient when at normal work. Since in stern gear it is only the outer portion which is effective, it is desirable that the tips be made fairly broad.

Turbine Machinery in its various forms is now common in express steamers of all sizes and services, as well as in the ships of the British and some foreign navies, where it has quite displaced the reciprocating engine. The Parsons turbine has long been a favourite, and the success of this instrument, both on land and shipboard, is due so very largely to the genius of that gentleman that it will ever be associated with his name. The Curtis, Zoelly, Rateau, and other turbines are fitted to ships with results equalling, if not excelling, those already recorded by the Parsons, the Brown-Curtis being very efficient.

The turbine must have, of necessity for high efficiency, a great peripheral speed; the diameter of rotor or the revolutions must, therefore, be great, but for very large power direct-driven the latter cannot be. In the case of the s.s. "Lusitania" (fig. 39), the L.P. rotor is 140 inches diameter, with the last set of blades 22 inches long, and running on bearings 33 inches diameter at 194 revolutions per minute on trial, and about 185 on service, and weighing 120 tons. For smaller powers the rate of revolution is much higher, and in case of a warship of 5,000 H.P. per shaft the rate of revolution is as much as 500, and higher, even up to 700 revolutions, in smaller ships of 4,750 H.P. per shaft. Such revolutions necessitate not only a screw of small diameter, but one of such very small pitch that the pitch ratio is so low that the efficiency of the propeller is lower than in competing ships. The slip ratio, however, is wonderfully small in the turbine-driven ship, taking all these things into account, being only 15·3 per cent. in the "Lusitania," and seldom over 25 per cent. in others where revolution speed is not very excessive. Formerly Sir C. Parsons fitted more than one screw to each shaft, but without the success he anticipated; the German Admiralty also tried the same method for improving the propeller efficiency of the turbine-driven ship, "Lubeck," with the same disappointment; consequently for such a ship a single screw of moderately small diameter and large disc ratio is the rule. That is, the screw is somewhat larger than would be fitted, if the regard of the designer were limited to turbine efficiency only. As a matter of fact, in all cases of this kind the choice of screw and all that pertains to it is governed by combining the efficiency of propeller and generator, as will be seen later on.* The turbine, when of a power exceeding 1,000 S.H.P., is superior to the best reciprocator in steam consumption per unit of power; the turbine of any size has a higher mechanical efficiency than a reciprocator of equal power; it occupies about the same amount of floor space as the ordinary triple and quadruple engine, but is of less height, so that much of it can go under deck, consequently the engine hatches can be very much smaller for them than the reciprocator. The consumption of lubricants is less and fewer attendants are required in the engine-room when on service. The difference in weight is trifling, but the prime cost and the repair and wear and tear account of the modern make of turbine compares favourably with that of the average reciprocator. On the other hand, the steam efficiency of the turbine falls off on reduction of load, and since the marine turbine suffers reduction in velocity when the speed of the ship is reduced, the fall in efficiency is considerable; so much so, indeed, that in the case of the "Lusitania," where the consumption of steam per S.H.P. per hour of the turbines alone was only 12·77 lbs. at the full speed of 25·4 knots per hour, it was as high as 21·23 lbs. at 15·77 knots. The corresponding coal consumptions, which, however, of course included that due to the requirements of the auxiliary machinery, were 1·46 lbs. and 2·76 lbs. respectively per S.H.P per hour. With the reciprocator there is no such rapid increase in coal consumption at the lower powers; on the contrary, the rate is lower at moderately reduced speeds than at full, and at quite low speeds is not very high. For example, on the trials of H.M.S. "Achilles" the steam consumption at full speed of 23·275 knots, with the engines developing 24,000 I.H.P., was 19·9 lbs. per I.H.P. per hour, and the coal 2·03 lbs., while at 14·6 knots it was only 16·95 lbs. of steam and 1·88 lbs. of coal; moreover, at 21·58 knots and 16,000 I.H.P. the coal consumption

* The employment of gearing has solved the problem of screw and turbine efficiency (*vide* Appendix A).

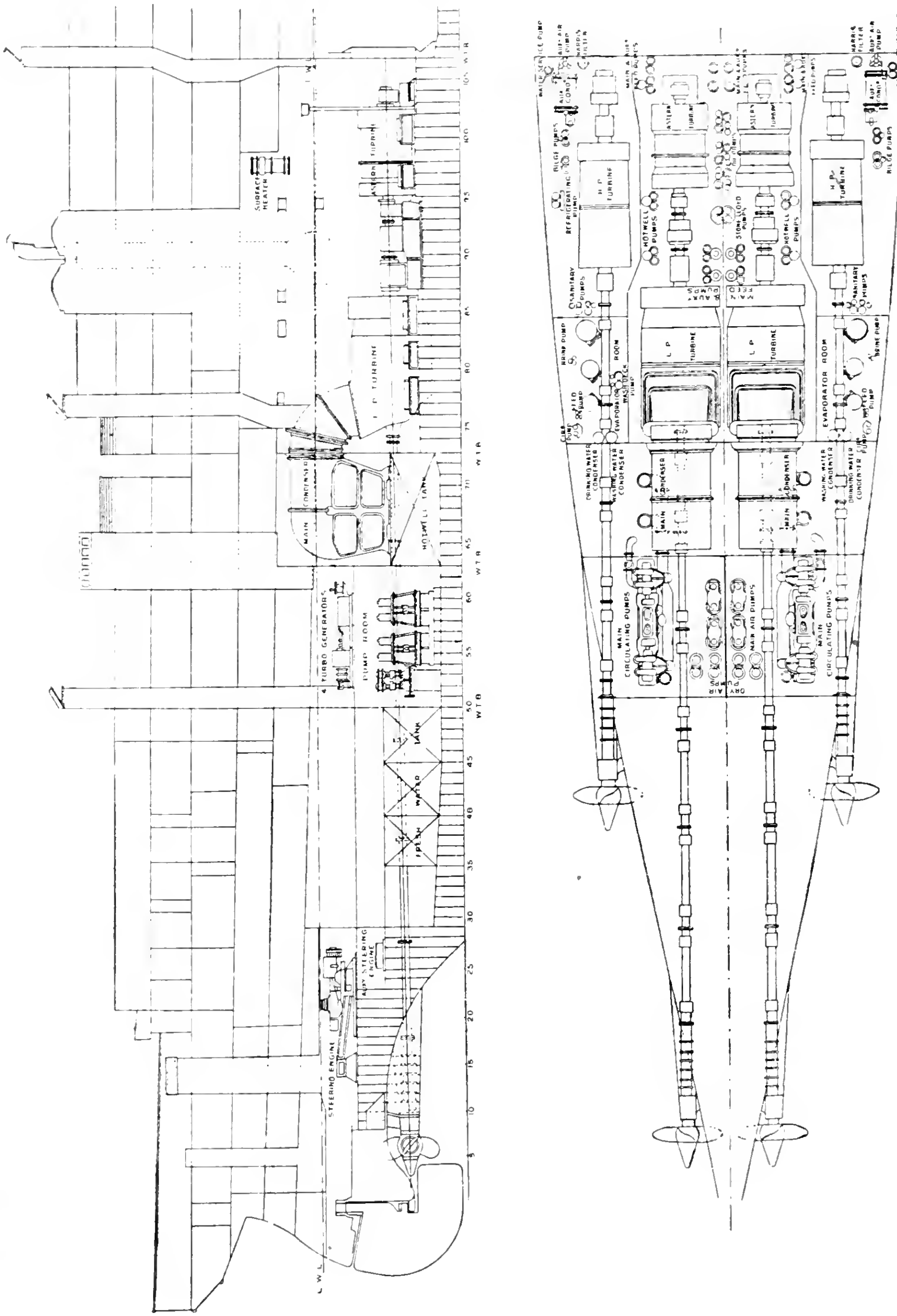


Fig. 39.—General Arrangement of Turbines and Shafting of R.M.S. "Lusitania." 76,000 S.H.P

was 1·85 lbs. only. Her engines were four-cylinder four-crank triple-expansion reciprocators.

The Admiralty, to test the Turbine, caused to be carried out some exhaustive comparative trials with H.M.S. "Amethyst," fitted with Parsons turbines, and her sister ship, the "Topaze," having the four-cylinder triple-compound reciprocators. These ships are each 360 feet long, 40 feet beam, and 14·5 feet mean draught of water; their displacement is 3,000 tons, and wetted skin about 16,100 square feet, with block coefficient of fineness of 0·503. Their boilers were similar in all respects, and their trials were conducted on exactly

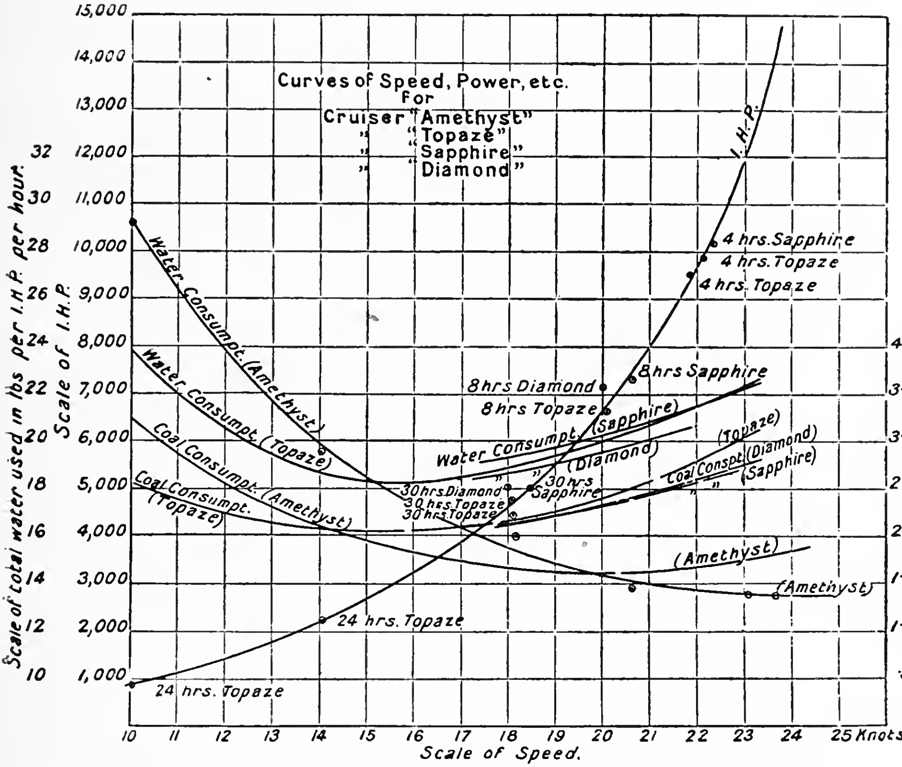


Fig. 40.—Trials of "Amethyst" (Turbines) and Sister Ships (Reciprocators).

the same lines, with the result that, whereas when the "Topaze" was exerting the maximum power of her reciprocating engines, 9,868 I.H.P., the speed attained was 22·103 knots, the turbines of the "Amethyst" developed a S.H.P. equivalent to about 14,200 I.H.P., which drove her at a speed of 23·63 knots. Fig. 40 gives the curves of power consumption, etc., of the two ships and their sister ships fitted with reciprocators.

The Experiments made in the U.S. of America with the cruiser "Birmingham," fitted with triple-compound reciprocators, and the "Salem," having Curtis turbines, is equally interesting. These ships are otherwise

TABLE XIV*a*.—Comparative Trials of Ships fitted with Turbines and Reciprocating Engines.

Name,	S.S. Lusitania.		H.M.S. Achilles.		H.M.S. Amethyst.		H.M.S. Topaze.		U.S.A. Salem.		U.S.A. Birmingham.	
Length, ft.,	760·0		480		360		360		420		420	
Beam, ft.,	87·5		73·5		40·0		40·0		47·0		47·0	
Draft water, mean, ft.,	32·5		26·6		14·6		14·6		16·75		16·75	
Displacement, tons,	36,440		13,000		3,000		3,000		3,750		3,750	
Wetted skin, sq. ft.,	82,360		39,200		16,100		16,100		22,000		22,000	
Coefficient fineness,	0·609		0·507		0·503		0·503		0·50		0·50	
Trial Speed,	Full.	Slow.	Full.	Slow.	Full.	Slow.	Full.	Slow.	Full.	Slow.	Full.	Slow.
Speed, knots,	25·4	15·77	23·27	14·60	23·63	14·06	22·10	14·08	25·95	11·93	24·33	12·23
Shaft horse-power,	68,850	13,400	..	..	..	..	..	..	19,200	1,360	..	..
Indicated horse-power,	..	..	23,970	4,880	14,200	2,250	9,868	2,254	..	..	15,540	1,600
Revolutions,	190	116	139·4	84·75	490	290	246	150	378	..	192	91·4
$S^3 \times D^{\frac{2}{3}} \div$ H.P.,	278	322	298	362	193	257	228	258	256	349	259	321
Total steam consumption, lbs.,	14·94	28·2	19·9	16·95	13·42	19·56	20·18	18·77	..	..	..	..
Coal consumption, lbs.,	1·46	2·76	2·03	1·88	1·74	2·10	2·65	2·06	1·81	2·68	1·92	2·89
Engine,	Turbine (Parsons)		4-cylr. triples		Turbines (Parsons)		4-cylr. triples		Turbine (Curtis)		Reciprocators.	
No. of screws,	Four		Two		Three		Two		Four		Two	
Boiler press,	150		200	185	260	263	250	198	..	..	..	..

alike, and 420 feet long, 47 feet beam, 16·75 mean draft, the displacement 4,700 tons, and the wetted skin about 22,000 square feet. They are of very fine form, the block coefficient being 0·50. The Curtis turbines drove the "Salem" at a speed of 25·947 knots, with 19,200 S.H.P., and a coal consumption of 2·01 lbs. per S.H.P. per hour (equivalent to 1·81 per I.H.P. of a reciprocator). The reciprocators drove the "Birmingham" at 24·325 knots speed, with 15,540 I.H.P., consuming 1·92 lbs. of coal; at 12·23 knots with 1,600 I.H.P., the consumption of coal of the "Birmingham" was 2·89 lbs., against the 2·68 lbs. of the "Salem" at 11·93 knots with 1,360 S.H.P.

Experiments were made by the German Government with the cruiser "Lubeck," of 3,170 tons displacement, 341 feet long, 43 feet beam, and 16·4 feet draft of water, driven by turbines operating on four shafts, each shaft having two screws at a speed of 23 knots, and the twin-screw sister ship, "Hamburg," having reciprocating engines; but in this case the superiority of the turbine was not demonstrated, inasmuch as it took 14,158 H.P. to attain a speed of 23·16 knots with the "Lubeck," and only 11,582 for 23·17 knots with the reciprocators. Doubtless, however, this was in no small measure due to the screw arrangement, which was a bad one, and probably the propeller efficiency was very low. Since those experiments were made, the German Admiralty have followed the lead of the British, and now are having turbines of kinds fitted in their warships.

Table XIV*a*. summarises the above.

The following table is interesting and instructive, showing, as it does, the comparative steam consumptions of the "Amethyst" and "Topaze" at speeds varying from 10 knots to 22, including the steam used by the auxiliaries. It must be borne in mind, when considering the same, that the engines of the "Topaze" were not designed primarily for economy, consequently their consumption of steam per I.H.P. is very high compared with that of the quadruple-compound engines of the mercantile marine, such as fitted in s.s. "Saxonia," constructed by John Brown & Co., and tested by the Admiralty Boiler Committee, where it was found to be 14·5 lbs. per I.H.P. when on service. At the same time, the thermal efficiency of the "Topaze's" engines was by no means bad considering the conditions under which they worked, for her steam consumption in main engines only was but 16·91 at a speed of 20 knots and 15·45 at 18 knots:—

TABLE XV.—WATER CONSUMPTION PER I.H.P. PER HOUR OF H.M.S. "AMETHYST" AND "TOPAZE" ON PROGRESSIVE TRIALS.

Speed in Knots.	10	11	12	13	14	15	16	17	18	19	20	21	22
H.M.S. "Amethyst" (Turbs.),	29·3	26·0	23·7	22·0	20·4	19·0	17·9	16·8	15·9	15·2	14·7	14·3	14·0
H.M.S. "Topaze" (Recipros.),	23·8	22·0	20·6	19·6	18·8	18·4	18·3	18·4	18·7	19·2	19·8	20·5	21·4

Combination of Turbines with Reciprocators seems to be the most likely development of marine steam machinery in the future, and it has already been adopted successfully by Denny & Co., also by Harland & Wolff in the Transatlantic steamships recently built by them for the White Star Company and others. Since the triple-expansion engine is more economic than is the boiler end half of a turbine, while the other or condenser end of the latter is much more economic than the reciprocator, and can make good use of a high vacuum, such a combination of the two is obviously a fitting one, and a desirable thing. Further, since at slow and cruising speeds the turbine is not economical, while the triple-compound engine is very fairly so, these latter engines can be employed by themselves to propel without using the turbines by exhausting direct to the condenser. Fig. 41 shows an arrangement proposed by Parsons for dealing with a three-screw ship; here each wing screw is driven by a four-cylinder triple-expansion engine arranged to exhaust either to its own condenser or to a low-pressure turbine operating a central screw and exhausting to the same condenser. Fig. 42 is an example by the same gentleman for a four-screw arrangement, in which each inner screw has its own triple-compound engine exhausting direct to its own condenser, or to a low-pressure turbine operating a wing screw, and exhausting to the condenser. That is, in each case a turbine is interposed between the L.P. cylinder and the condenser, in order that the steam from it at 10 lbs. pressure absolute may be usefully employed in expanding down to the pound or even less pressure of the condenser.

Messrs. Denny Brothers, of Dumbarton, fitted the s.s. "Otaki" with triples and low-pressure turbines, and demonstrated that the gain over the arrangement with triple engines in the sister ship, s.s. "Orari," was as much as 17 per cent. Careful experiments with triple-expansion engines at electric power stations on shore show that fully 15 per cent. more power is obtained with the same consumption of steam if a low-pressure turbine is interposed in this way between it and the condenser.

The following are the figures given by Com. Wisnom, of Denny's, from the trials of the above two steamers :—

The Performance of s.s. "Otaki," having two sets of ordinary triple-expansion engines, each driving a wing propeller as in a twin-screw ship, and both exhausting to a low-pressure turbine driving a propeller on the middle line as in a single-screw ship; built by Denny Bros., of Dumbarton, for the New Zealand S. Co., and sister ship to the twin-screw steamer "Orari," which was, however, 4·5 feet shorter.

The "Otaki" has a displacement of 9,900 tons on 27·5 mean draught, and her principal dimensions—

Length between perpendiculars, . . .	464·5 feet.
Beam moulded, ...	60·0 ,,
Depth ,, ...	34·0 ,,

Each of her reciprocating engines has cylinders $\frac{24{\cdot}5''\text{-}39''\text{-}58''}{39''}$, while each of those of the sister ships, "Opawa" and "Orari," are $\frac{24{\cdot}5''\text{-}41{\cdot}3''\text{-}69''}{48''}$. The turbines of the "Otaki" has a rotor 90 inches diameter.

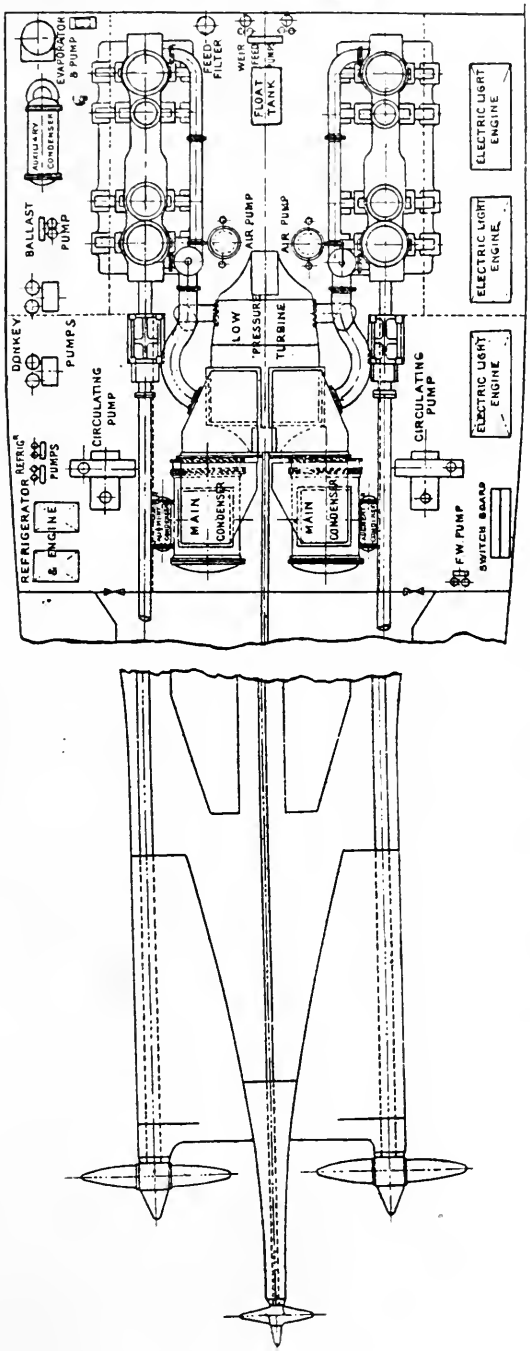


Fig. 41.—Combination of a Central Turbine with Twin Reciprocators (Parsons).

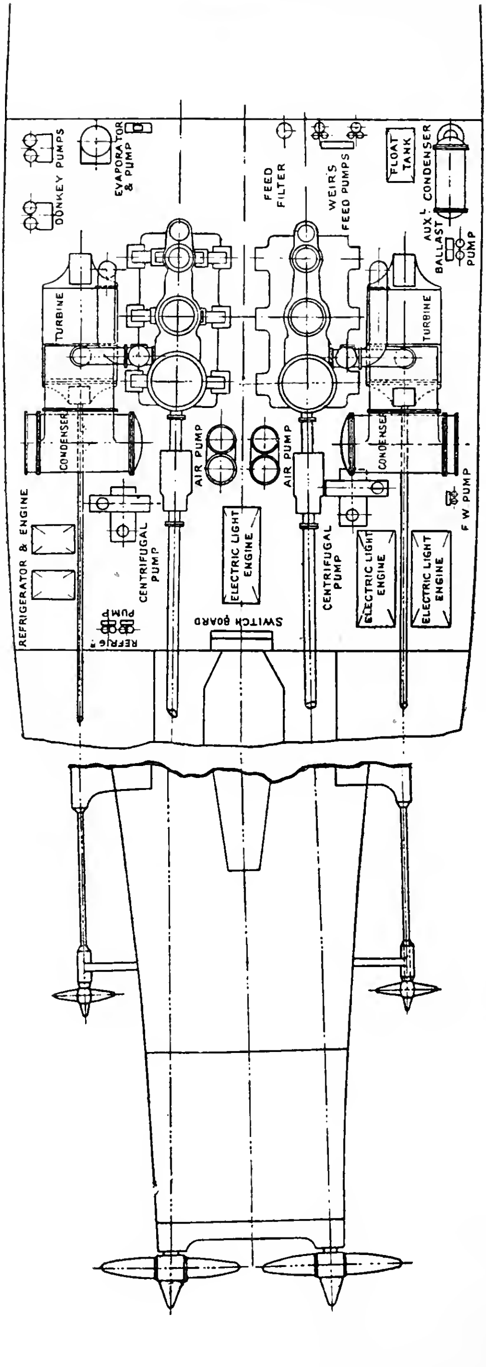


Fig. 42.—Combination of Twin Turbines with Twin Reciprocators (Parsons).

The following is a comparative summary of results of trials at 14·6 knots speed on the measured mile :—

Name of Ship.	E.H.P.	I.H.P.	Propulsive Coefficient	Water Consumption per Hour.		
				Total.	Per E.H.P.	Per I.H.P.
			Per cent.			
3-screw s.s. "Otaki" (turbo-recipro.),	3,350	5,880	57	73,300	21·9	13·7
2-screw s.s. "Orari" (recipros.),	3,210	5,360	60	88,300	27·5	16·5
Gain per cent. in "Otaki,"	..	..	..	17	20	17

TABLE XVI.—Measured Mile Trials of s.s. "Otaki," October 31, 1908, on 20 Feet Mean Draught.

	Mean of A Runs.	Mean of B Runs.	Mean of C Runs.	Mean of D Runs.
Total horse-power, being I.H.P. (recipros.) + S.H.P. (turb.),	6,857	5,348	4,704	3,282
Mean speed, knots,	15·02	14·28	13·83	12·52
Revolutions, recipros.,	103·5	97·9	93·5	83·4
,, turbine,	224·5	209·7	197·2	172·1
Total water consumption per hour, lbs.,	82,000	67,300	60,200	44,600
,, ,, ,, per H.P. ,,	11·95	12·6	12·8	13·6
Mean absolute pressure at H.P. cylinder, ,,	193	178	166	135
,, ,, turbine inlet, ,,	9·5	7·62	6·76	5·0
Vacuum at exhaust end of turbine,	28·1	28·2	28·4	28·5
,, on condenser gauge,	28·2	28·4	28·3	28·5
Temperature of sea water, F.°	56	56	56	56
,, circulating discharge,	70	67	70	70
,, hot well,	72	70	73	74
Steam consumption based on the I.H.P. of s.s. "Orari" by tanks,	13·66	13·7	13·8	13·07
As measured by pumps per I.H.P. per hour,	14·12	14·1	14·3	15·2

Figs. 43 and 44 show the general arrangement of the machinery of the s.s. "Olympic," built by Harland & Wolff for express service between England and New York. She is one of the largest ships, and has worked with satisfactory results; she is 860 ft. long, 92·75 ft. beam, and 32·5 draft of water, displacement 50,000 tons, and I.H.P. 54,800; propelled by three screws, the two wing ones worked by reciprocating triple-compound engines, each having four cylinders, 54, 84, 97, and 97 inches diameter, and 63 inches stroke, and the middle by low-pressure Curtis turbine, taking the steam supply from the L.P. cylinders of the reciprocators (*v.* fig. 44).

A Development with existing Single-screw Ships could be made by fitting two low-pressure turbines abaft the old triple engine, each operating a wing screw, and both exhausting to the old condenser; this would probably

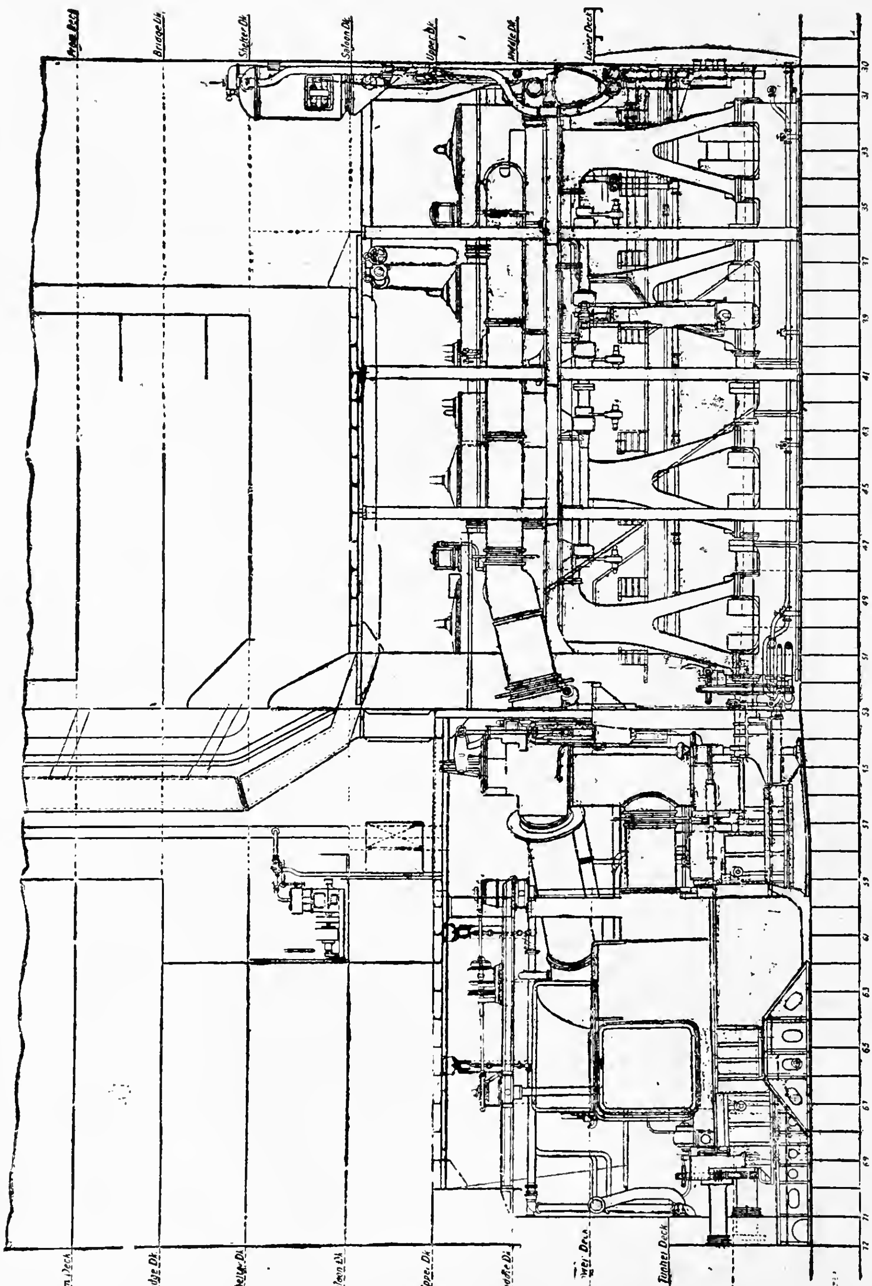


Fig. 43.—Longitudinal Section through Engine-Room of R.M.S. "Olympic," built by Harland & Wolff, Belfast.

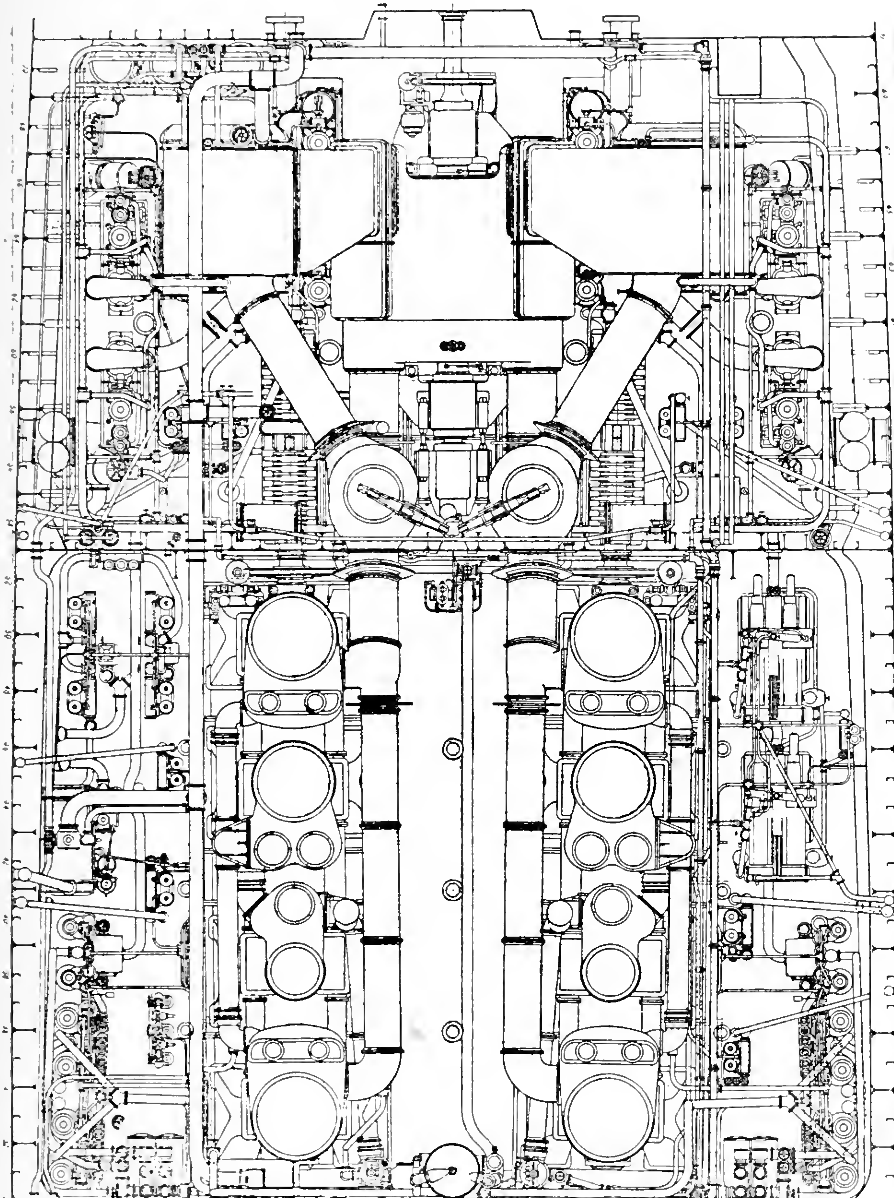


Fig. 44.—Plan of Engine-Room of R.M.S. "Olympic," 50,000 tons displacement. Two Triple-Expansion Reciprocators and One L.P. Turbine, 55,000 H.P.

be quite as good for cargo and "mixed" steamers as pulling out the old engines and fitting two turbines geared to the original screw shafting (fig. 45), as done by Sir C. Parsons in the s.s. "Vespasian." Fig. 46 shows such an arrangement proposed by this gentleman; it is one quite easily carried out, and although it may be questionable if it is worth doing to an ordinary cargo boat, it certainly would be quite a good thing for very many of the combined cargo and passenger steamers designed for long voyages at speeds from 12 to 15 knots with single screws. The gain of 13·7 per cent. in power shown by him (in the schedule later on) is very material, and the alternative 15 per cent. of fuel important in all places, but very much so where coal is 20s. to 30s. per ton delivered on board the ship in ordinary times.

Oil Engines of moderate power are usually of the well-known single-acting type (figs. 17 and 49), having trunk pistons connected direct to the crank-pins

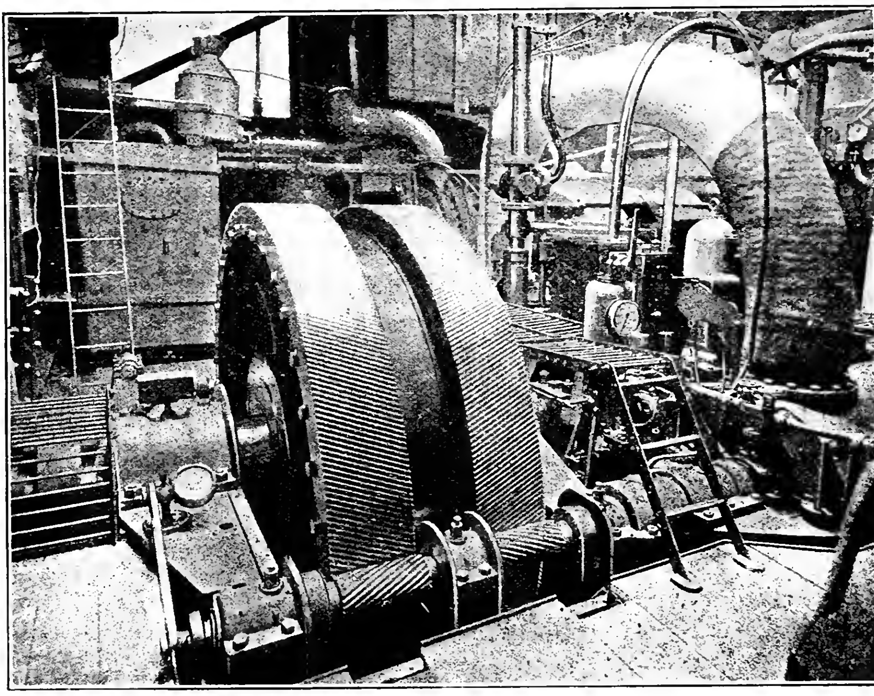


Fig. 45.—Wheel-Gearing of s.s. "Vespasian" (Parsons).

by ordinary connecting-rods and valve-gear wheel driven; they seem to work very well at steady loads, and are handled by compressed air. Increases of power are obtained by increasing the numbers of cylinders, so that it is not at all uncommon to find eight cylinders in line operating on one line of shafting with a screw at the outer end. There is, of course, in this way no limit to the number of cylinders, and probably the cost of increase by this method is no greater than if the greater power were obtained with cylinders of large size; sometimes cylinders are placed in tandem axially—that is, in rear of or above each cylinder is another operating with it on the same connecting-rod.

Trouble, however, is experienced sometimes with these trunk pistons when of large size, so that recourse is had now to the piston-rod type when the cylinders increase in diameter ; at present 30 inches diameter * is considered about as large as should be made for marine engines on the Diesel system, where the initial pressure is exceedingly high, especially if compared with the mean. It is sometimes as high as 40, but is generally 35 atmospheres in these engines at commencement of the stroke, but, owing to the high compression effected, there is little or no shock at commencement of stroke. This engine, however, notwithstanding such initial pressures, has become popular in this country, as it had been for some time on the Continent, where it has proved successful when working with that comparatively safe fuel heavy oil with a high flash point or residuals.

The Design of Oil Engines was at first very similar to that followed for land engines. Now, however, the tendency is to conform to the marine practice found to be the best for steam engines ; their builders also adopt the enclosed type with forced lubrication, and so obtain good and safe running at high rates of revolution. Fig. 17 is a good example of a Thornycroft 100 H.P. marine petrol engine, the reversing of which is made by means of a clutch gear, which does well enough in small craft, being similar to the method by which motor-driven vehicles are reversed.

The Diesel Engine is generally designed to work on the usual four-cycle system, and its *modus operandi* is as already described (*v.* figs. 50 and 50*a*), but attached to it is a three-stage air compressor, which not only supplies

* *Vide* Appendix A

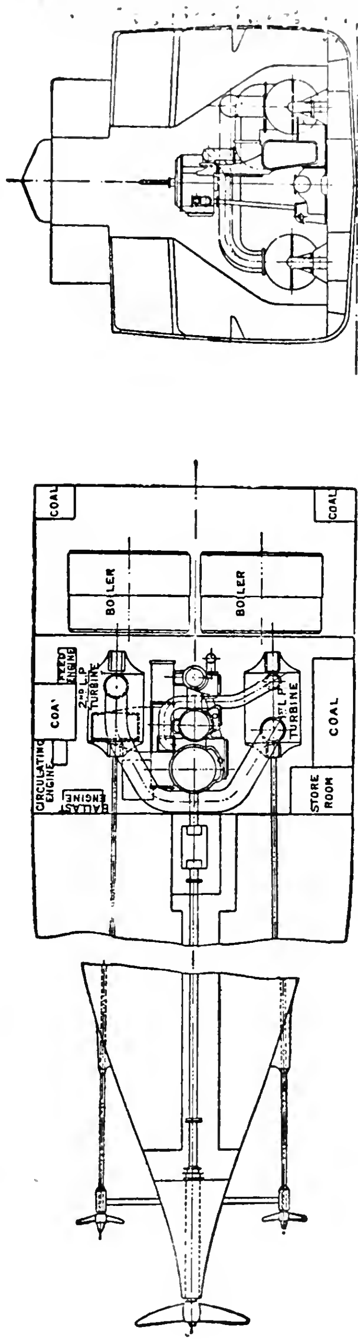


Fig 16 —Combination of a Triple-Expansion Engine with Two Low-pressure Turbines in Series (Parsons).

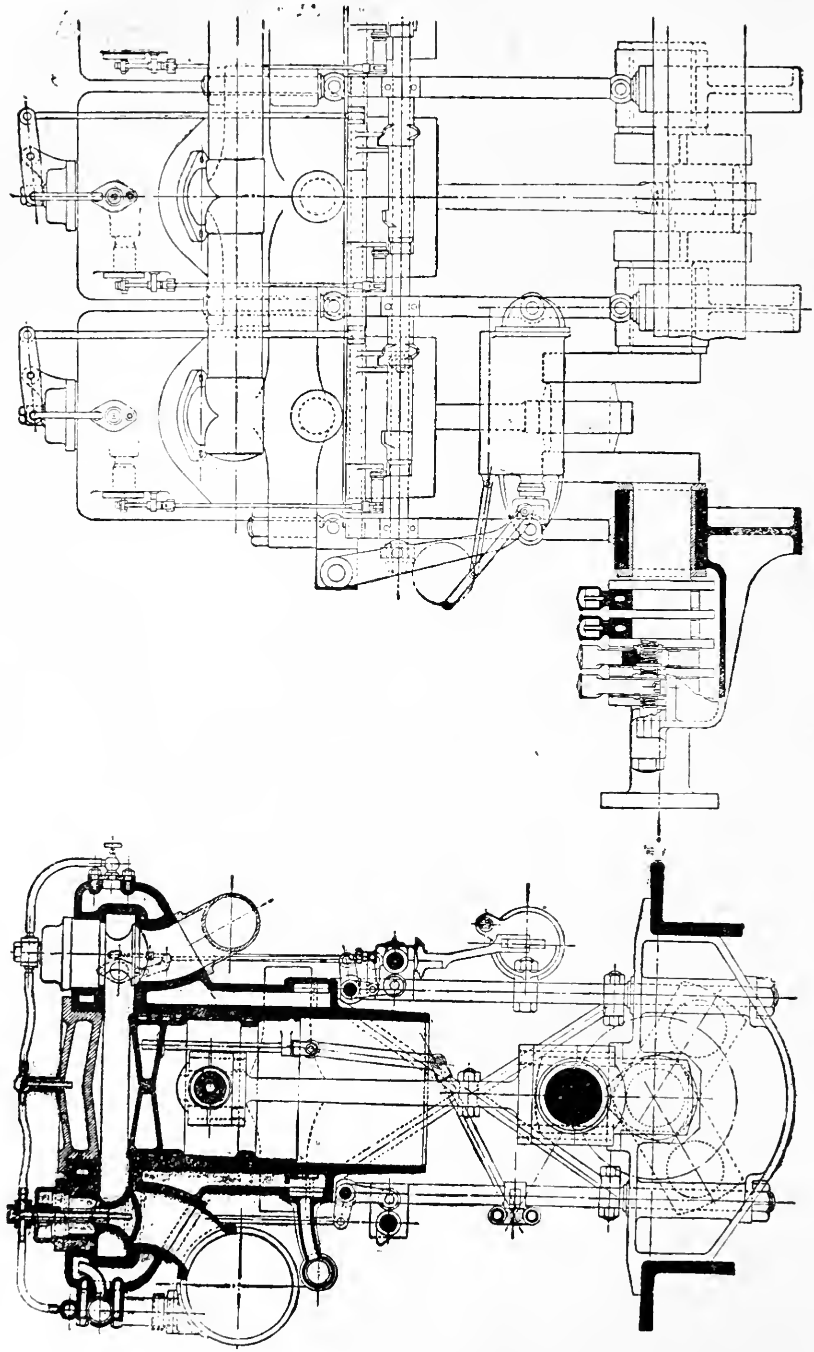


Fig. 47.—Oil Engine, Single-Acting Marine Design. Six Cylinders, 12½ ins. × 14 ins.

the air for injecting the oil, but provides a means of working the engine and reversing it when required by using the oil cylinders as in an air engine, when, by special means, it is put out of action as an oil engine. The engines on the two-cycle system have also one or more low-pressure pumps (*v.* fig. 50*a*) to supply the air for scavenging and filling the cylinders. In some marine designs these pumps are worked by levers, and like the ordinary air pumps of a steam engine. The high rate of compression of the air prevents shock on exploding the mixture of air and oil vapour, as would be the case if the load came on suddenly, as it does in the ordinary engine when there is no such compression. The ratio of maximum to mean pressure (4·31) is exceedingly high in all such engines, and consequently the rods, framing, crank shafts, etc., must be large by comparison with those of steam engines, it follows that the weight of these engines per horse-power is very great, and not much less than that of a steam installation including the boilers; on the other hand, the consumption of oil fuel is only about 40 per cent. that in an oil-fired steamship.

The double-acting cylinder in which explosives take place on both sides of the piston have yet to be proved equal to continuous service; doubtless the pistons must be in that case water-cooled, and the stuffing-boxes most carefully made and maintained to be satisfactory. To compete with large powers the double-acting engine is desirable, but it still remains for engineering skill and resource to get over these practical difficulties. In time, too, the reversing may not be effected by quite such clumsy and indirect means as prevail at present with gearing or subsidiary compressed air arrangements. Fig. 49 is a sectional view of a Diesel engine as made by Mirrlees Bickerton Company for small power on the four-cycle system, and non-reversible.

Double-acting Diesel two-cycle engines of considerable power have been made in large sizes on the Continent. Here the cycle with its compression and combustion follows its course as in the single-acting engine, but in this case on both sides of the piston, as in fig. 48. The power developed is thus practically doubled in a given size of cylinder; or the same power obtained with about half the cylinder capacity. There are the usual practical objections to this system of overheating both pistons and cylinders, although they are water-cooled, and the difficulty with stuffing-boxes exposed to such high temperatures. Moreover, to obtain such high compression the clearance must be very small, for with 35 atmospheres at *each* end it must be less than 3 per cent. of the cylinder capacity, and consequently the stroke clearance about 2 per cent.—that is, with an engine of 10 inches stroke it is not more than $\frac{3}{10}$ inch at each end, and must not vary materially from this at any time.

The stuffing-box difficulty may be overcome by having hollow rods, through which water is passed to the pistons, and kept in circulation as Dr. Kirk did with steam for heating the L.P. pistons of the early compound engines made by J. Elder & Co. for the Navy. It is, however, very doubtful if the two-cycle double-acting engine will be satisfactory for even intermittent running, and there is reason to think it is unlikely to be so for continuous work on shipboard in large sizes.

The Two-cycle Diesel Engine differs from the four inasmuch as the fresh air is admitted above the piston when it is at the bottom of the stroke

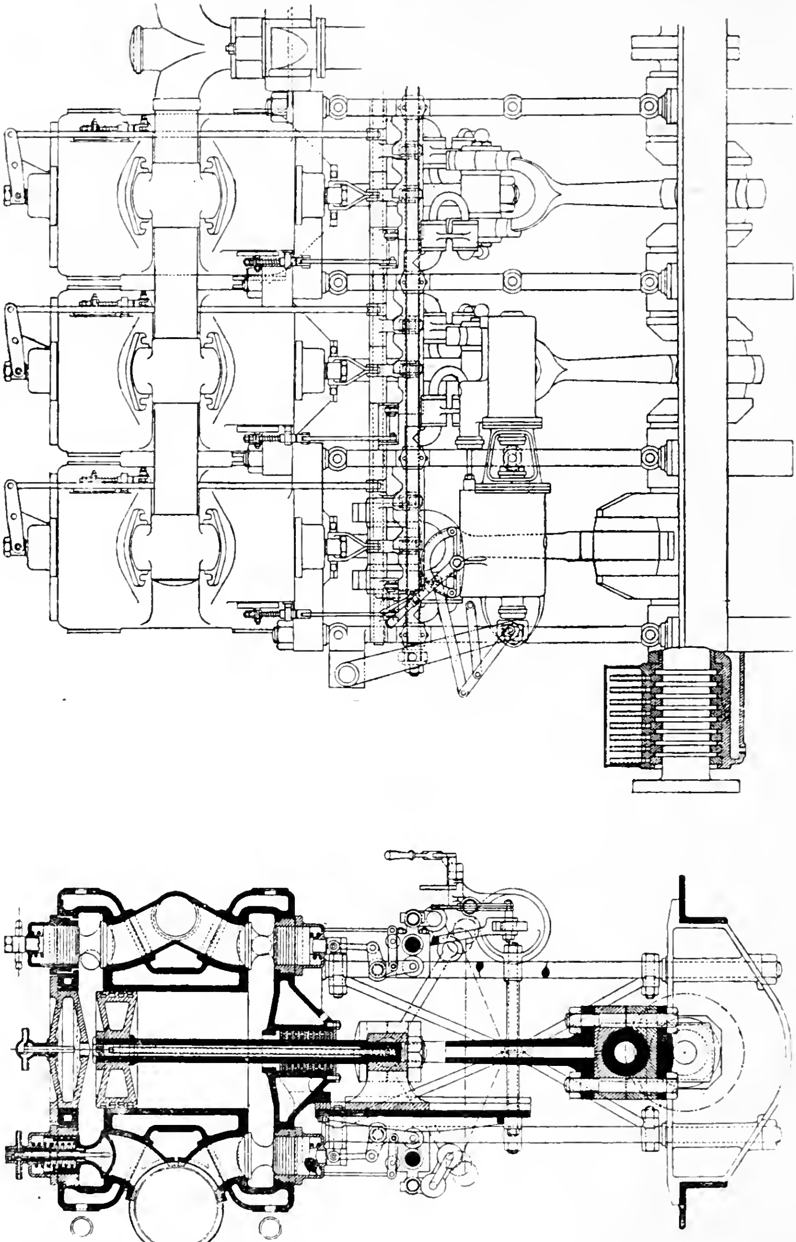


Fig. 48.—Oil Engine, Double-Acting, Reversible. (Cylinders, 10 inches by 10 inches).

after combustion, the products having begun to escape at six-sevenths the stroke are by it ejected and replaced, the scavenging orifice closing again one-seventh from the bottom; the piston continuing its stroke compresses it to the end when the oil is sprayed in as before. In this way an impulse is made at every revolution, instead of at every other one as in the "Otto" cycle. This still further increases the power developed by a unit of cylinder capacity, but it likewise increases the heat production and the difficulties that arise from high temperatures.

The Fuel Consumption of these engines is generally less than 0·4 pound of oil per horse-power, and in very favourable cases as low as 0·348 lb. Taking 10 lbs. of steam as a very good performance for a turbine, and 16 lbs.

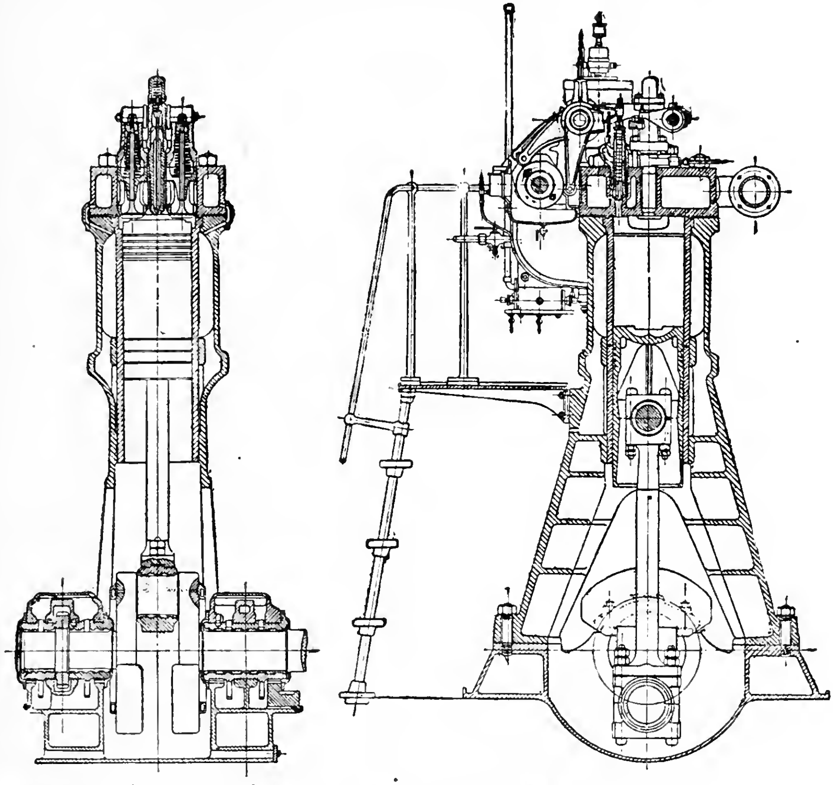


Fig. 49.—Single-Acting "Mirrlees-Diesel" Marine Oil Engine.

of steam per pound of oil when burned to be produced in a good boiler, then

Consumption of oil per hour of a turbine = 10 ÷ 16, or 0·625 lb.

If the consumption of the "Lusitania" be taken as 12·77 lbs. in the turbines alone, and 14·46 lbs. the total for all purposes, the oil fuel consumption will be 0·798 and 0·903 lb. per S.H.P., equal to 0·766 and 0·867 per I.H.P. respectively.

Taking, however, the consumption of the "Otaki" with the combined turbo-reciprocators as 12 lbs. of steam per hour, the oil fuel for her would be 0·75 lb. Although this consumption of oil fuel is considerably greater than that of the oil engine, the amount of lubricating oil in the Diesel is

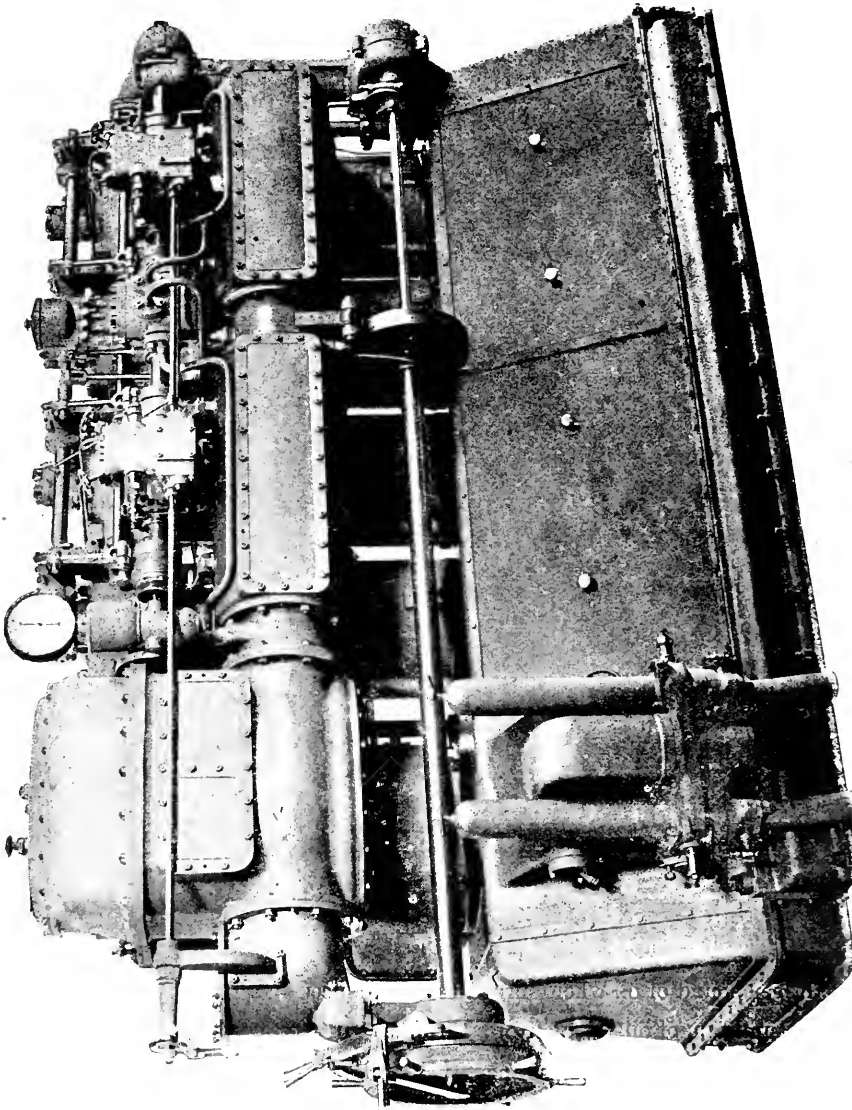


Fig. 50.—Marine Oil Engine, 400 B.H.P. Two-Cycle Single-Acting on Diesel System.

very much higher; in fact, excessive compared with that of a reciprocating steam engine of equal power.

When oil engines are used on board sea-going ships, it is necessary for them to have power for steering and other purposes generated by an independent oil engine and electrically distributed. The whistle may be blown

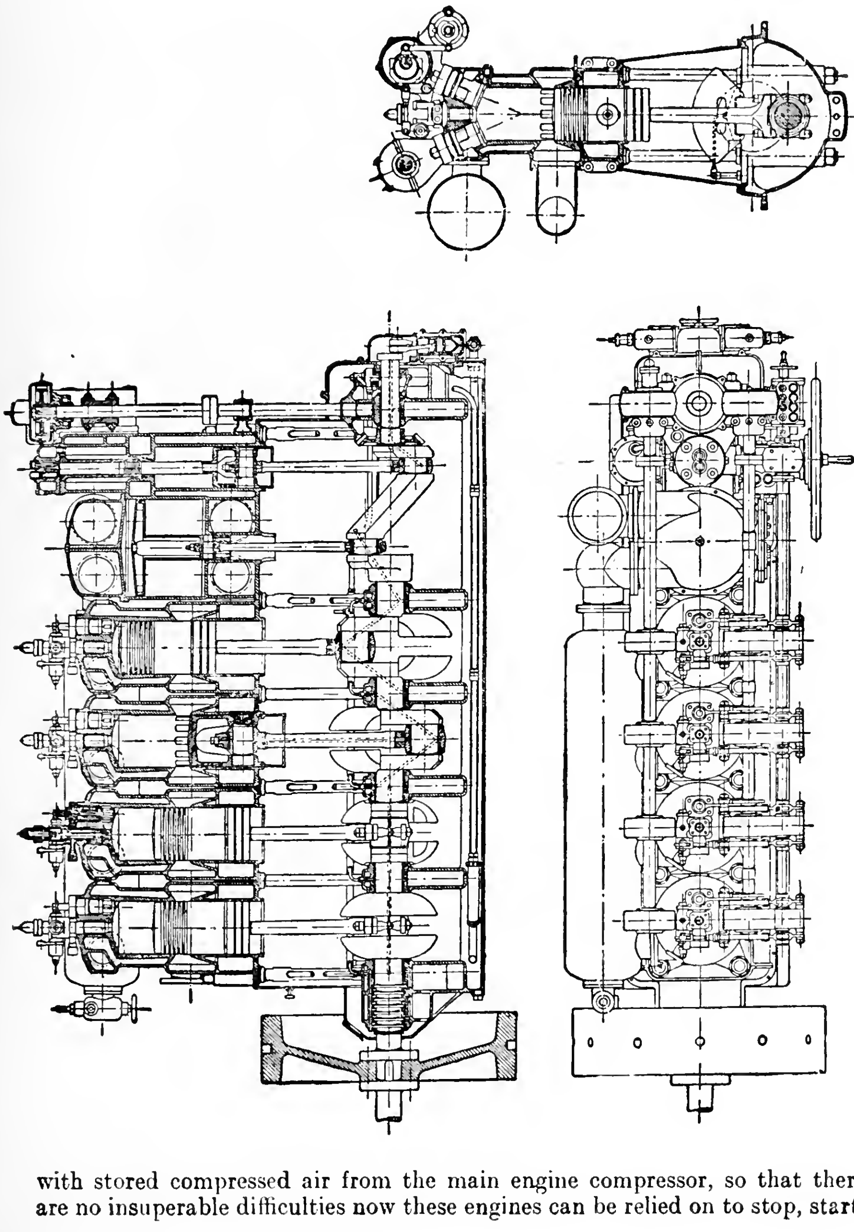
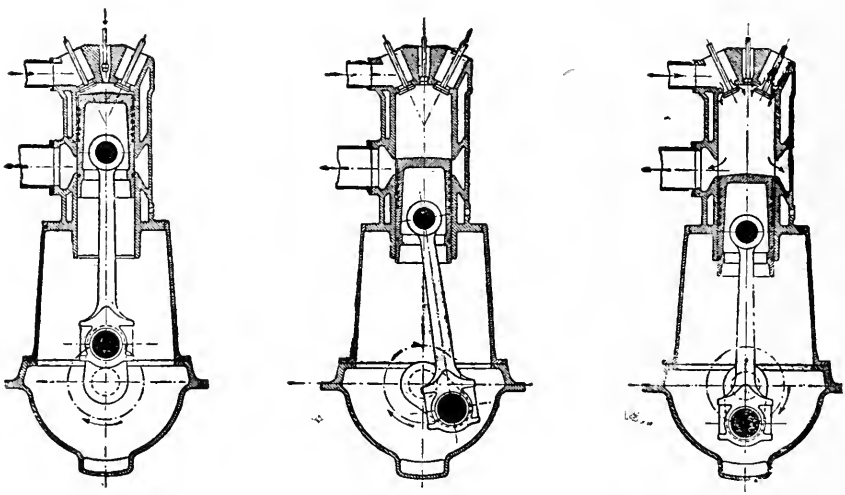


Fig. 50*a*.—Marine Heavy Oil Two-Cycle Engine, 400 B.H.P., showing Air Compressors and Reversing Gear.

with stored compressed air from the main engine compressor, so that there are no insuperable difficulties now these engines can be relied on to stop, start, and reverse quickly, and to run for a week on end without stopping to clean carburetters, etc., and certainly without stopping unexpectedly at awkward times from this latter cause. Mr. Westgarth thought it prudent to provide a small steam generator for such purposes above-named in the ship fitted by him with oil engines, so that there might not be so many novelties at one time.

Gas Engines on board ship were pretty much of the same general design as the oil engine, but they require a gas producer, etc., to supply them with fuel, which adds to their weight and the space occupied by them. Moreover, as Prof. Vivian Lewis very properly pointed out, "the suction plant suffers from the limitation that before it can achieve commercial success afloat a form of generator and scrubber, occupying small space, must be devised, in which bituminous coal can be used as the fuel to be gasefied, and the gas supplied freed from all tar vapour. . . . I am not aware



1. Charged with Compressed Air, and Fuel being admitted.

2. Expansion completed and Piston about to open Exhaust.

3. End of Down Stroke. Scavenge Valve open. Clearing out.

Fig. 51.—Two-Cycle Oil Engine (Diesel System), showing the Three Stages.

that it has yet been successfully done. The mechanical troubles of caking and arching of the fuel in the generator can be overcome, but many years' experience of efforts to decompose or get rid of tar vapour has impressed me with a great respect for the difficulties of the problem and a perfectly clean gas, absolutely free from tar vapour, is the first essential for success with the gas engine."

All this is very true, though it must be a matter of regret, seeing that this country abounds in coal, but has very little oil. For this reason alone the adoption of the oil engine in British waters is scarcely politic, either in warships or cargo-boats.

CHAPTER IV.

STEAM USED EXPANSIVELY.

In the Reciprocating engine work is done by the elastic force of the steam acting on the pistons, and *pressing* them forward on their strokes against the back pressure behind them during its expansion from the time it enters the cylinder to the time it is allowed to escape to the free atmosphere in the case of a non-condensing or to the condenser of a condensing engine.

From the time of entering to the time of cut-off expansion is taking place, though it is slight and often not appreciable; after cut-off it is real, considerable, and quick; it is continuous to the end, and the rate is expressed by the ratio of the capacity of the L.P. cylinder to that portion of the H.P. which was filled with steam at cut-off. This is nominally the rate, and would be really so if the admission valve were large and wide open till it closed and cut-off made suddenly, as is the case with the Corliss or "drop" valves. In practice, however, at the exact point of cut-off with the ordinary slide valves the steam is wire-drawn down considerably below the pressure in the valve chamber, which latter may be taken as the initial pressure.

With Steam Turbines steam is admitted continuously instead of intermittently as in reciprocators; the expansion commences immediately it enters the nozzles or their equivalents, and may be wholly carried out in one nozzle, and the whole velocity due to the fall from boiler pressure to exhaust pressure generated at once, and the kinetic energy expended on the blades of a single rotor, as shown in fig. 52. The well-known De Laval turbine is on this principle, and consequently runs at a very high rate of revolution, since the velocity acquired by steam in falling from 160 lbs. to 3 lbs. absolute (vacuum 24 inches) is 3,660 feet per second, and the peripheral velocity of rotor for this flow, if the efficiency is to be good, must be not much less than 1,800 feet; for a turbine of this kind with a rotor 38 inches diameter, the rate of revolution should be 180 per second, or 10,800 per minute; as a matter of fact, a De Laval turbine of 300 H.P. has a rotor 30 inches diameter running at 10,600 revolutions per minute.

The Expansion may be in Stages, however, and fig. 52*a* shows how this may be accomplished in a rudimentary way by causing the first rotor to run in a chamber where the pressure is below that at entry, but is somewhat above that at the final exhaust to condenser. The velocity of flow will then be much less than that stated, as the drop will be less (say to 25 lbs. absolute instead of 3). The steam from that chamber will pass into and through another nozzle, where further expansion takes place with a renewal of velocity to the steam, thereby giving it further kinetic energy to be expended on the blades of a second rotor. In all cases of turbines, the rate of expansion is expressed by the ratio of the initial pressure to that at exhaust—that is, $p_1 \div p_o = r$.

In the above example $r = 160 \div 3$, or 53·3.

Moderately Moist Steam expands in accordance with Boyle and Marriott's law—viz., whereby the pressure varies inversely as the volume—that is, $pv = c$.

Then,

$$\text{The mean pressure} = p_1 \frac{1 + \text{hyp. log } r}{r}.$$

The hyperbolic logarithm of a number may be found by multiplying the common logarithm of that number by 2·302585.

There is a simplicity in this rule that commends it to the practical mind, and as steam in marine engines is usually fairly moist, it may generally be used to solve with sufficient accuracy the every-day problems connected

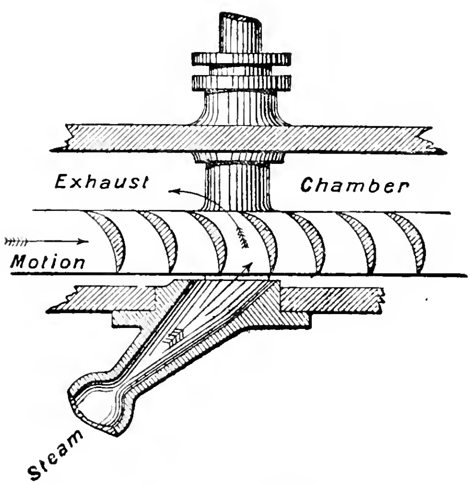


Fig. 52.—Turbo-Motor (De Laval system).

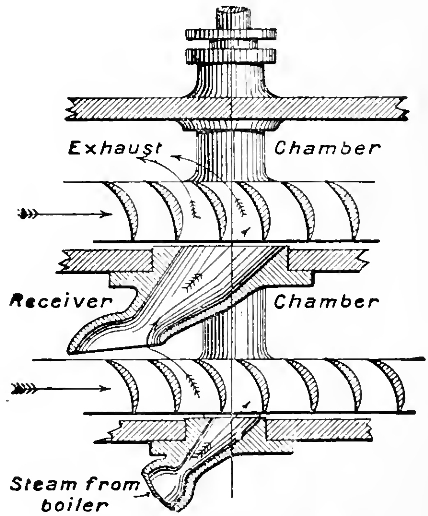


Fig. 52a.—Compound Turbo-Motor (De Laval system).

with the marine engine. Such an expansion at constant temperature is called isothermal.

$$\text{The terminal pressure} = \frac{p_1}{r}.$$

Therefore,

(1) Ratio of mean pressure to terminal pressure $= \dfrac{rp_m}{p_1}$.

(2) Ratio of terminal pressure to mean pressure $= \dfrac{p_1}{rp_m}$.

(3) Ratio of maximum pressure to mean pressure $= \dfrac{p_1}{p_m}$.

(4) Dry steam is assumed to expand $pv^{1\frac{1}{6}}$ = constant.

The following table will be found useful, and contains the multipliers for ascertaining the mean pressure of steam when expanding on Boyle's law, as well as when expanding adiabatically.

TABLE XVII.—Steam used Expansively.

r	$\frac{1}{r}$	$\frac{rp_m}{p_1}$	$\frac{p_1}{rp_m}$	$\frac{p_1}{p_m}$	$\frac{p_m}{p_1}$	$\frac{p_m}{p_1}$ Dry.
20	0·050	4·00	0·250	5·00	0·1998	0·186
18	0·055	3·89	0·256	4·63	0·2161	0·200
16	0·062	3·77	0·265	4·24	0·2358	0·220
15	0·066	3·71	0·269	4·05	0·2472	0·230
14	0·071	3·64	0·275	3·85	0·2599	0·242
13·33	0·075	3·59	0·279	3·72	0·2690	0·254
13	0·077	3·56	0·280	3·65	0·2742	0·258
12	0·083	3·48	0·287	3·44	0·2904	0·271
11	0·091	3·40	0·294	3·24	0·3089	0·292
10	0·100	3·30	0·303	3·03	0·3303	0·314
9	0·111	3·20	0·312	2·81	0·3552	0·340
8	0·125	3·08	0·321	2·60	0·3849	0·370
7	0·143	2·95	0·339	2·37	0·4210	0·408
6·66	0·150	2·90	0·345	2·30	0·4347	0·417
6·00	0·166	2·79	0·360	2·15	0·4653	0·450
5·71	0·175	2·74	0·364	2·08	0·4807	0·466
5·00	0·200	2·61	0·383	1·92	0·5218	0·506
4·44	0·225	2·50	0·400	1·78	0·5608	0·540
4·00	0·250	2·39	0·419	1·68	0·5965	0·582
3·63	0·275	2·29	0·437	1·58	0·6308	0·616
3·33	0·300	2·20	0·454	1·51	0·6615	0·648
3·00	0·333	2·10	0·476	1·43	0·6993	0·670
2·86	0·350	2·05	0·488	1·39	0·7171	0·707
2·66	0·375	1·98	0·505	1·34	0·7440	0·733
2·50	0·400	1·91	0·523	1·31	0·7664	0·756
2·22	0·450	1·80	0·556	1·24	0·8095	0·800
2·00	0·500	1·69	0·591	1·18	0·8465	0·840
1·82	0·550	1·60	0·626	1·14	0·8786	0·874
1·66	0·600	1·51	0·662	1·10	0·9066	0·900
1·60	0·625	1·47	0·680	1·09	0·9187	0·913
1·54	0·650	1·43	0·699	1·07	0·9292	0·926
1·48	0·675	1·39	0·718	1·06	0·9405	0·940

When steam expands in accordance with the law $p\,v$ = constant, the curve drawn through the extremities of ordinates representing the pressure at any position of the piston is a hyperbola. The mean height of such a system of ordinates is found by the formula given above; this mean height will represent the mean pressure.

The mean pressure may be obtained without the aid of logarithms, by resorting to arithmetical calculation of the ordinates, and finding the mean by the method usually followed with indicator diagrams.

Example.—Initial pressure 80 lbs., rate of expansion 5. Suppose the length of stroke divided into ten equal parts by points 1, 2, 3, . . . 9, 10. The cut-off is $\frac{1}{5}$, or two-tenths the stroke.

The pressure at commence of stroke is				80 lbs.
„	1 tenth	„		80 „
„	2 „	„		80 „
„	3 „	„	$\frac{2}{3}$ of 80 or	53·33 „
„	4 „	„	$\frac{2}{4}$ „	40·00 „
„	5 „	„	$\frac{2}{5}$ „	32·00 „
„	6 „	„	$\frac{2}{6}$ „	26·66 „
„	7 „	„	$\frac{2}{7}$ „	22·86 „
„	8 „	„	$\frac{2}{8}$ „	20·00 „
„	9 „	„	$\frac{2}{9}$ „	17·78 „
„	10 „	„	$\frac{2}{10}$ „	16·00 „

$$p_m = \frac{1}{10}\left(\frac{80 + 16}{2} + 80 + 80 + 53{\cdot}33 + 40 + 32 + 26{\cdot}66 + 22{\cdot}86 + 20 + 17{\cdot}78\right)$$
$$= 42{\cdot}063 \text{ lbs.}$$

By reference to Table xvii., with 5 for the rate of expansion, $\frac{p_m}{p^1} = 0{\cdot}5218$, and for 80 lbs. pressure $p_m = 41{\cdot}744$ lbs., or about 0·75 per cent. less than that given by the summation above, the excess of which is due to the small number of points of observed pressure.

Graphic Method.—Professor Rankine proposed * a geometric method of ascertaining the mean pressure, which is of interest and value, and which first appeared in the *Engineer*. Draw a straight line, A B, of definite length, produce it to A C, so that $A\,C = \frac{A\,B}{4}$. Through A draw A D at right angles to C A B. With C as centre, and C B as radius, draw the arc of a circle, cutting A D at D. Then if $\frac{D\,A}{D\,E}$ is the rate of expansion $\frac{p_m}{p_1} = \frac{E\,F}{A\,B}$.

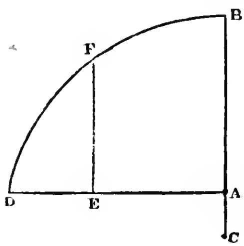


Fig. 53.

To suit this diagram for actual use, A B should be taken of such a length as is convenient for scale measurement, say 4 inches; A D should be divided into ten parts and subdivided into quarters; through the divisions faint lines should be drawn parallel to A B. Scales should be constructed 4 inches long, suitable for the usual pressures coming under consideration.

The object of columns 3, 4, in Table xvii., is that mean pressure and initial pressure may be easily determined from terminal pressures, when the rate of expansion is known; or when initial and mean pressures are known, the rate of expansion may be found. Column 5 is given to show the relation between maximum and mean pressures at the various rates of expansion.

Adiabatic Expansion.—When steam is dried by slight superheating, so that it is surcharged with heat, and is capable of very considerable expansion without liquefaction taking place, it expands according to the law of perfect gases, and then

* Rankine, *Rules and Tables*, p. 291.

$$\frac{p_m}{p_1} = \frac{17 - 16r^{-\frac{1}{16}}}{r}$$

$r^{-\frac{1}{16}}$ may be found by extracting the square root of $\frac{1}{r}$ four times.

$$\left(r^{-\frac{1}{16}} = \sqrt{\sqrt{\sqrt{\sqrt{\frac{1}{r}}}}}\right)$$

Column 6 gives the value of $\frac{p_m}{p_1}$ as calculated from the above formula. It will be seen that, except at very high rates of expansion, there is no very great difference between the ratio as calculated by this method and by the method for moderately moist steam.

Clearance.—In practice the mean pressure in the cylinder is very materially affected by what is called *clearance*. However accurately the engine is constructed, there is always at the commencement of the stroke a space between the piston and cut-off valve, made up of the part of the cylinder between the piston and the cover or cylinder end, and the passage between valve face and cylinder; this is called the *clearance*. Supposing this space is equal to one-tenth of the capacity of the cylinder, and the cut-off is at two-tenths the stroke, the *effective* cut-off is not two-tenths, but something more, due to the fact that the expansion of a volume of steam equaling three-tenths the capacity of cylinder is being effected, instead of that of a volume of two-tenths. This practically amounts to making the cylinder 10 per cent. longer, and cutting off at three-tenths the stroke without clearance. It is, therefore, customary to speak of the clearance as equal to a certain fraction of the stroke. This, however, must be distinguished from the lineal clearance or distance of the piston from the cylinder ends while at the extreme limits of its stroke. This should be expressed always as a definite length, and not as a fraction of the stroke; it may be, and often is, called *stroke clearance*, while the space is *volume clearance*.

To allow for the effect which the *clearance* will have when steam expands in a cylinder, let r be the nominal rate of expansion as before, and r_1 be the *actual* rate allowing for clearance, c the clearance as a fraction of the cylinder capacity. Then

$$\frac{1}{r_1} = \frac{\frac{1}{r} + c}{1 + c} \quad \text{and } r_1 = r\,\frac{1 + c}{1 + cr} \quad \cdot \quad \cdot \quad \cdot \quad \text{(A)}$$

$\frac{1}{r} + c$ being the volume of steam at cut-off between the piston and the cut-off valve, and which expands to the volume $1 + c$ at the end of the stroke. If there is no cushioning of the steam before admission, then the whole of the space $\frac{1}{r} + c$ must be filled at each stroke with fresh steam.

Then the *real mean absolute pressure* will be

$$p'_m - c(p_1 - p'_m) \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \text{(B)}$$

p' is the mean pressure obtained by means of Table xvii., the *actual* rate of expansion being taken, and p_1 is the absolute initial pressure. If, however, there is sufficient cushioning to fill the clearance space with steam at the initial pressure, the volume of steam used at each stroke will be only that swept by the piston at cut-off and equal to $\frac{1}{r}$.

Compression or Cushioning.—There will be an increase of back pressure caused by this cushioning, and its effect on the mean pressure is as follows:—

Let p'_m be the mean absolute pressure due to the *effective* cut-off $\frac{1}{r_1}$; p_0 the absolute back pressure; c the clearance; and p_1 the absolute initial pressure as before.

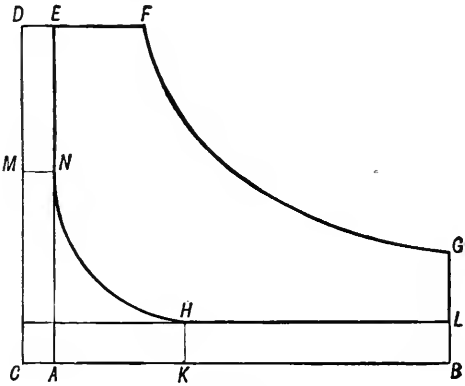


Fig. 54.

Fig. 54 is the indicator-diagram of an engine working under these conditions. A B is the stroke, A C the clearance, E F the nominal cut-off, and D F the effective. The actual rate of expansion is therefore $\frac{\text{C B}}{\text{D F}}$ C D represents the initial pressure and H K the back pressure. Cushioning commences at K with pressure p_0, and at A the pressure is p_1.

The figure C D E H K is the diagram due to the expansion of steam of a volume equalling c at a pressure p_1 to a pressure p_0, so that the rate of its expansion is $\frac{\text{C K}}{\text{D E}}$. Now $p_1 \times \text{D E} = p_0 \times \text{C K}$, and, therefore,

$$\frac{p_1}{p_0} = \frac{\text{C K}}{\text{D E}}.$$

Since p_1 and p_0 are both known the rate of expansion is known, and by referring to Table xvii. the mean pressure p''_m due to this rate of expansion is found.

The area H E F G $=$ area C D F G B $-$ (area C D E H K $+$ area H K B)

„ C D F G B $= p'_m \times (\text{A B} + \text{A C}) = p'_m (1 + c)$.

„ H K B $= p_0 (\text{A B} - \text{A K}) = p_0 \left(1 - (c\frac{p_1}{p_0} - c)\right)$.

„ C D E H K $= p''_m (\text{C K}) = p''_m \left(\frac{p_1}{p_0} c\right)$.

„ H E F G $= p_m \times 1$, or the effective mean pressure.

Therefore, the effective mean pressure

$$= p'_m (1 + c) - \left\{1 - c\left(\frac{p_1}{p_0} - 1\right)\right\} p_0 - p''_m \left(\frac{p_1}{p_0} - c\right)$$

$$= p'_m \times c\,(p'_m - p_0) + cp_1 \left(1 - \frac{p''_m}{p_0}\right) - p_0$$

$$= (p'_m - p_0)(1 + c) + cp_1 \left(1 - \frac{p''_m}{p_0}\right). \quad - \quad - \quad - \quad (C)$$

General Effect of Clearance and Cushioning.—Let p', the absolute initial pressure, be represented in Fig. 54 by C D, p_0 the back pressure by B L, A B the length of stroke, A C the clearance c, A K the compression x, E F the nominal cut-off, r the nominal rate of expansion, r_1 the real rate of expansion, &c., &c., as before. p'_m the mean pressure due to an initial pressure p', and a rate of expansion r_1; p_m the real mean pressure of N E F G L H. As before

$$r_1 = \frac{1 + c}{\frac{1}{r} + c} = r\frac{1 + c}{1 + cr}. \quad - \quad - \quad - \quad (1)$$

Since the steam at point K is shut up in a space $x + c$, and is compressed into a space c, the *rate of compression* is $\frac{x + c}{c}$; and the pressure after compression at N is $p^c = \frac{x + c}{c} p_0$, and represented by A N or C M; let p^c_m be the mean pressure of the figure M N H K C, which is that due to a pressure $\frac{x + c}{c} p_0$, and a rate of expansion $\frac{x + c}{c}$.

The area NEFGLH = CDFGB - DENM - MNHKC - KHLB.

„ NEFGLH $= p_m \times 1$.

„ CDFGB $= p'_m (1 + c)$.

„ DENM $= \left(p' - p_0 \frac{x + c}{c}\right) c = (p' - p_0)\,c - p_0 x$.

„ MNHKC $= p^c_m (x + c)$.

„ KHLB $= p_0 (1 - x)$.

Therefore,

$$p_m = p'_m (1 + c) - \{(p' - p_0)\,c - p_0 x\} - p^c_m (x + c) - p_0 (1 - x)$$

$$= p'_m (1 + c) - p'c - p_0 (1 - 2x - c) - p^c_m (x + c). \quad - \quad (2)$$

Example I.—To find the effective mean pressure in a cylinder having a clearance space equal to one-seventh its capacity, the initial pressure 80 lbs. absolute, the back pressure 10 lbs. absolute, and the nominal cut-off $\frac{1}{5}$ the stroke:

(1) If no compression

$$r = 5\,\frac{1 + \frac{1}{7}}{1 + \frac{5}{7}} = 5 \times \tfrac{8}{12} = 3{\cdot}33.$$

By reference to Table xvii. $\frac{p_m}{p_1} = 0{\cdot}6615$ for a rate of expansion $= 3{\cdot}33$

Then

$$p'_m = 80 \times 0{\cdot}6615 = 52{\cdot}92 \text{ lbs.},$$

and

The effective mean pressure = 52·92 - 10 = 42·92 lbs.

(2) If full compression to 80 lbs.: Here rate of compression

$$r_0 = \frac{p_1}{p_0} = \frac{80}{10} = 8.$$

Therefore,

$$\frac{p^c_m}{p_0} = \frac{r_0 p^c_m}{p_1} = 3{\cdot}08 \text{ (Table xvii.)}$$

$$p'_m = 52{\cdot}92 \text{ lbs. as before.}$$

Then the effective mean pressure by formula (C), p. 115,

$$= (52{\cdot}92 - 10)(1 + \tfrac{1}{7}) + \tfrac{80}{7}(1 - 3{\cdot}08).$$

$$= \tfrac{8}{7} \times 42{\cdot}92 - \frac{166{\cdot}4}{7} = 25{\cdot}28 \text{ lbs.}$$

If there was no clearance, the effective mean pressure would be 41·74 - 10, or 31·74 lbs.

The steam used in the case (2) is the same as if there had been no clearance, and as the effective mean pressure was only 25·28 lbs., there is a *loss due to clearance* of 6·46 lbs., or 20 per cent. In case (1) the quantity of steam used is $\frac{12}{35}$ the volume of the cylinder per stroke, or one-seventh of the volume in excess of the quantity with no clearance, so that with this increase of steam, if there was no clearance and the rate of expansion 5, there should be an increase in the work done, and that increased work will be to the work done by the smaller quantity of steam as 12 is to 7.

The equivalent mean effective pressure is then $\frac{12}{7}$ of 31·74, or 54·41, as against 42·92 lbs. which was obtained, showing a loss of 11·49 lbs., or 21 per cent.

The example given is a very extreme case, and such as would be rarely found in practice. The effect of clearance in the high-pressure cylinder of a compound engine may be seen in the following:—

Example II.—The nominal rate of expansion is 2, the initial absolute pressure 90 lbs., and the absolute back pressure $22\frac{1}{2}$ lbs., the clearance being one-ninth the capacity of the cylinder.

(1) No compression:

$$r = 2\frac{1 + \frac{1}{9}}{1 + \frac{2}{9}} = 1{\cdot}82.$$

By reference to Table ix., $\frac{p_m}{p_1}$ = ·8786 for a rate of expansion of 1·82.

Then $\quad p_m = 90 \times 0{\cdot}876 = 79$ lbs.

Mean effective pressure = 79 - 22·5, or 56·5 lbs.

The equivalent mean pressure when $\frac{1}{2} + \frac{1}{9}$ or $\frac{11}{18}$ of the volume of the cylinder of steam is used will be $\frac{11}{9}$ of 53·68 lbs., or 65·61 lbs., showing a loss by clearance of 13·88 per cent.

(2) If full compression to 90 lbs.:

Here $\frac{p_1}{p_0} = 4$, which is the *rate of compression;* so that

$$\frac{p^c_m}{p} = \frac{r_0 p^c_m}{p_1} = 2{\cdot}39. \quad \text{Table xvii.}$$

The effective mean pressure by formula (C), p. 115.

$$= (79 - 22{\cdot}5)\,(1 + \tfrac{1}{9}) + \tfrac{90}{9}\,(1 - 2{\cdot}39)$$
$$= 62{\cdot}77 - 13{\cdot}9 = 48{\cdot}87 \text{ lbs.}$$

Thus showing a loss of 5·81 lbs., or 10·8 per cent. only. The loss from the clearance in a compound is not so serious as in the expansive engine, and in the H.P. and M.P. cylinders of a triple or quadruple engine is of no consequence, as the steam in the former (which has passed from the high-pressure cylinder without giving out its full work), will do more work in the medium-pressure and low-pressure cylinders; whereas, with the expansive engine, the exhaust steam passes direct to the condenser at a higher pressure than if there is no clearance. Further, since the cut-off in an expansive engine is much earlier than in a compound, and the clearance from practical considerations is very much the same, the ratio of clearance to volume at cut-off will be much higher in the former than in the latter. Considerable loss is, however, experienced in compound, triple, and quadruple engines if the clearance in the low-pressure cylinder is large, therefore in that it should be as small as possible.

The beneficial effect of cushioning is seen in both the preceding examples, but its value is greater still when the cut-off in the high-pressure cylinder is somewhat earlier, as may be seen by the following:—

Example III.—The cut-off in the cylinder of example (2) is altered to $\frac{1}{4}$ the stroke, so that the nominal rate of expansion is 4.

(1) No compression:

$$r = 4\,\frac{1 + \frac{1}{9}}{1 + \frac{4}{9}} = 3.$$

Then $$p_m = 62{\cdot}1 \text{ lbs.}$$

and the effective mean pressure $= 62{\cdot}1 - 22{\cdot}5 = 29{\cdot}6$ lbs.

The equivalent mean pressure due to the amount of steam used is now $\frac{13}{9}$ of 31·185, or 45 lbs., thus showing a loss of 12 per cent.

(2) If full compression to 90 lbs.

The effective mean pressure by formula (C)

$$= (62{\cdot}1 - 22{\cdot}5)\,\tfrac{10}{9} + \tfrac{90}{9}\,(1 - 2{\cdot}39)$$
$$= 44 - 13{\cdot}9 = 30{\cdot}1 \text{ lbs.}$$

Thus showing a loss of 1·085 lbs., or 3·4 per cent. only.

The economy effected by working with a considerable amount of cushioning is, therefore, very appreciable, and experience has proved the correctness of this.

In actual practice, however, it only happens that so much cushioning can be effected as to fill the clearance space with steam of pressure equal to that entering in the H.P. cylinder of a triple or quadruple engine which exhausts steam of high pressure; but still even what is conveniently obtained materially adds to the economic working of the engine. It must not, however, be over looked that the *effective mean pressure* is considerably reduced by cushioning.

Mean Pressure in a Compound Engine.—If the effective mean pressure in the high-pressure cylinder of a compound engine be divided by the ratio of capacity of low-pressure to that of the high-pressure cylinder, the quotient represents the mean pressure necessary to do the same work in the low-pressure cylinder as is effected in the high-pressure cylinder. If this be added to the effective mean pressure in the low-pressure cylinder, the sum will be the mean pressure necessary to obtain from the low-pressure cylinder alone, the whole work done by both cylinders, and may be called the *equivalent mean pressure.* If there be no loss of mean pressure, owing to drop in the receiver, or other cause, this equivalent mean pressure will be the same as the effective mean pressure obtained by the steam expanding in one cylinder at the same rate as the total expansion effected in both cylinders of the compound engine. In the two-cylinder receiver form of compound engine, there is sometimes a considerable fall in pressure from the release point to the exhaust, owing to the low pressure maintained in the receiver, or to the late cut-off in the H.P. cylinder.

(1) Two-cylinder receiver compound engine.

Let p_1 be the initial pressure, p_0 the back pressure in the low-pressure cylinder, p_r the pressure in the receiver and back pressure in the high-pressure cylinder; R the ratio of cylinder capacities, r the total rate of expansion, r_1 the rate of expansion in the high-pressure cylinder, and r_n that in the low-pressure cylinder; p'_m the mean pressure due to an initial pressure, p^1, and a rate of expansion, r_1 : p''_m, the mean pressure due to an initial pressure p_r, and a rate of expansion r_n. P_m is the mean pressure due to a rate of expansion r, and an initial pressure p_1:

The effective mean pressure in the high-pressure cylinder is then $(p'_m - p_r)$; and that in the low-pressure cylinder is $(p''_m - p_0)$.

Also
$$P_m = p_1 \frac{1 + \text{hyp. log. } r}{r},$$

$$p'_m = p_1 \frac{1 + \text{hyp. log. } r_1}{r_1},$$

$$p''_m = p_r \frac{1 + \text{hyp. log. } r_2}{r_2}.$$

Since the work performed in the engine is supposed to be equally divided between the two cylinders,

$$p'_m - p_r = \text{R}\,(p''_m - p_0). \qquad (1)$$

But if there be no loss due to "drop," and the mean pressure in the high-pressure be referred to the low-pressure cylinder, then

$$\frac{p'_m - p_r}{\text{R}} + (p''_m - p_0) = P_m - p_0.$$

By substituting the value of $(p'_m - p_r)$ of (1) in the above

$$\left.\begin{aligned} p''_m - p_0 &= (P_m - p_0)\tfrac{1}{2} \\ \text{and} \quad p'_m - p_r &= (P_m - p_0)\frac{\text{R}}{2} \end{aligned}\right\} \qquad (2)$$

Let x be the efficiency of the system, so that $(1 - x)$ is the proportion of loss due to drop. Then

$$\left.\begin{aligned} p''_m - p_0 &= x\,(P_m - p_0)\tfrac{1}{2} \\ \text{and} \quad p'_m - p_r &= x\,(P_m - p_0)\frac{\text{R}}{2} \end{aligned}\right\} \qquad (3)$$

To find the *actual* mean pressures when there is loss due to "drop," the value of x must be determined; this may be done by substituting the value of p'_m and p''_m, found by the preceding formulæ; but an approximate value may be found by determining the value of p_r in equation (3); from the value thus found calculate p''_m, and refer the mean pressures of both cylinders to the low-pressure cylinder. If $(P'_m - p_0)$ be the equivalent mean pressure thus found, then, approximately,

$$x = \frac{P'_m - p_0}{P_m - p_0}. \qquad (4)$$

Example.—To find the mean pressure in a compound engine using steam of 90 lbs. absolute pressure, the total rate of expansion being 7, the ratio of the cylinder capacities 3·5, and the back pressure 4 lbs.

$$P_m = 90 \times 0{\cdot}421 = 37{\cdot}89 \text{ lbs.} \qquad \text{Table xvii.}$$

$$r_1 = 7 \div 3{\cdot}5 = 2.$$

Then

$$p'_m = 90 \times 0{\cdot}8465 = 76{\cdot}18 \text{ lbs.} \qquad \text{Table xvii.}$$

$$p_r = 76{\cdot}18 - \frac{3{\cdot}5}{2}(37{\cdot}89 - 4) = 16{\cdot}88 \text{ lbs.}$$

$$r_2 = \frac{p_r r}{p_1} = \frac{16{\cdot}88 \times 7}{90} = 1{\cdot}313,$$

$$p''_m = 16{\cdot}88 \frac{1 + \text{hyp. log. } 1{\cdot}313}{1{\cdot}313} = 16{\cdot}36.$$

That is, the effective mean pressure in high-pressure cylinder is 76·18 − 16·88, or 59·3 lbs., and that in low-pressure cylinder is 16·36 − 4, or 12·36. Referred to low-pressure cylinder alone,

$$P'_m - p_0 = 12{\cdot}36 + \frac{59{\cdot}3}{3{\cdot}5} = 29{\cdot}3 \text{ lbs.}$$

$$P_m - p_0 = 37{\cdot}89 - 4 = 33{\cdot}89 \text{ lbs.}$$

Therefore,

$$x = \frac{29{\cdot}3}{33{\cdot}89} = 0{\cdot}865.$$

Then, actual effective mean pressure in high-pressure cylinders $\Big\} = 0{\cdot}865\,(37{\cdot}89 - 4)\frac{3{\cdot}5}{2} = 51{\cdot}3$ lbs.

Then, actual effective mean pressure in low-pressure cylinder $\Big\} = 0{\cdot}865\,(37{\cdot}89 - 4)\tfrac{1}{2} = 14{\cdot}65$ lbs.

And the actual pressure in receiver is then - - - - - - $\Big\} = 76{\cdot}18 - \frac{3{\cdot}5}{2}(29{\cdot}3) = 24{\cdot}88$ lbs.

(2) *The three-cylinder receiver compound engine, having two low-pressure cylinders.* Ratio of each low-pressure cylinder to the high-pressure is $\frac{R}{2}$.

In this case only one-third of the work is done in each cylinder. Then

$$p'_m - p_r = \frac{R}{2}(p''_m - p_0), \qquad (1)$$

and as

$$\frac{p'_m - p_r}{R} + (p''_m - p_0) = P_m - p_0.$$

Then

$$p''_m - p_0 = \tfrac{2}{3}(P_m - p_0)$$

and

$$p'_m - p_r = \frac{R}{3}(P_m - p_0) \qquad (2)$$

Also the actual values

$$\left.\begin{aligned} p''_m - p_0 &= \tfrac{2}{3}\,(P_m - p_0)\,x \\ p'_m - p_r &= \frac{R}{3}\,(P_m - p_0)\,x \end{aligned}\right\} \quad . \quad . \quad . \qquad (3)$$

Example.—To find the mean pressures in a three-cylinder compound engine (having two low-pressure cylinders), using steam of 90 lbs. absolute pressure, the total rate of expansion being seven, the ratio of the combined capacity of the low-pressure cylinder to that of the high-pressure being 3·5, and the back pressure 4 lbs.

$$P_m = 90 \times 0{\cdot}421 = 37{\cdot}89 \text{ lbs.},$$

$$r_1 = \frac{r}{R} = 7 \div 3{\cdot}5 = 2,$$

$$p'_m = 90 \times 0{\cdot}8465 = 76{\cdot}18 \text{ lbs.},$$

$$p_r = 76{\cdot}18 - \frac{3{\cdot}5}{3}(37{\cdot}89 - 4) = 36{\cdot}64 \text{ lbs.},$$

$$r_2 = \frac{p_r\,r}{p_1} = \frac{36{\cdot}64 \times 7}{90} = 2{\cdot}85,$$

$$p''_m = 36{\cdot}64\,\frac{1 + \text{hyp. log. } 2{\cdot}85}{2{\cdot}85} = 26{\cdot}4$$

Then the mean effective pressure in the high pressure cylinder - - - - - $= 76{\cdot}18 - 36{\cdot}64 = 39{\cdot}54$ lbs.

Then the mean effective pressure in the low pressure cylinder - - - - - $= 26{\cdot}4 - 4 = 22{\cdot}4$ lbs.

Then

$$P'_m - p_0 = 22{\cdot}4 + \frac{39{\cdot}54}{3{\cdot}5} = 32{\cdot}7 \text{ lbs.}$$

Then

$$x = 32{\cdot}7 \div 33{\cdot}89 = 0{\cdot}965.$$

Then actual effective mean pressure in high-pressure cylinder - - - $= \frac{3{\cdot}5}{3}(37{\cdot}89 - 4) \times 0{\cdot}965 = 38{\cdot}15$ lbs.

Then actual effective mean pressure in each low-pressure cylinder - - $= \frac{2}{3}(37{\cdot}89 - 4) \times 0{\cdot}965 = 21{\cdot}8$ lbs.

(3) ***The three-cylinder compound continuous expansion engine, having one high-pressure, one low-pressure, and one medium-pressure cylinder, generally called triple-compound.***

R is the ratio of low-pressure to high-pressure cylinder; R_1 the ratio of low-pressure to mean-pressure cylinder; p' the initial pressure, &c., as before.

p'_m the mean pressure due to expansion, r_1, and pressure p',
p''_m " " " r_2, " p'',
p'''_m " " " r_3, " p'''.

p'' is the pressure in the receiver between high-pressure and mean-pressure cylinders, p''' that in the receiver between mean-pressure and low-pressure cylinders.

Then effective mean pressure in high-pressure cylinder $= p'_m - p''$,
,, ,, mean-pressure ,, $= p''_m - p'''$,
,, ,, low-pressure ,, $= p'''_m - p_0$

Then, if there is no loss due to drop,

$$p'_m - p'' = R(p'''_m - p^0) \text{ and } p''_m - p''' = R_1(p'''_m - p^0). \quad - \quad (1)$$

But

$$P_m - p^0 = p'''_m - p^0 + \frac{p''_m - p'''}{R_1} + \frac{p'_m - p''}{R}.$$

Therefore

$$\left.\begin{aligned} p'''_m - p^0 &= \frac{P_m - p^0}{3} \\ p''_m - p''' &= \frac{R_1}{3}(P_m - p^0) \\ p'_m - p'' &= \frac{R}{3}(P_m - p^0) \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This is true when there is no loss from "drop," but, as in practice there is generally some loss from this cause, an approximation must be found in a similar way to that for the two-cylinder compound engine.

The cut-off in the high-pressure cylinder will be, as before,

$$\frac{1}{r_1} = \frac{R}{r}.$$

The cut-off in mean-pressure cylinder in order to maintain a pressure, p'', in the receiver between it and the high-pressure cylinder can be found in the same way as before. Since R is the ratio of low-pressure to high-pressure cylinder, and R_1 that of low-pressure to mean-pressure cylinder, $\frac{R}{R_1}$ will be the ratio of the mean-pressure to high-pressure cylinder, then

$$\frac{p''}{r_2} \times \frac{R}{R_1} = \frac{p'}{r_1},$$

and

$$\frac{1}{r_2} = \frac{R_1}{R} \times \frac{p'}{p''} \times \frac{1}{r_1}.$$

Substituting the value of $\frac{1}{r_1}$. Then, $\frac{1}{r_2} = R_1 \frac{p'}{p''r}$.

The cut-off in low-pressure cylinder to maintain a pressure, p''', in the receiver between it and the mean-pressure cylinder,

$$\frac{1}{r_3} = \frac{1}{R_1} \times \frac{p''}{p'''} \times \frac{1}{r_2}.$$

Substituting the value of $\frac{1}{r_2}$. Then, $\frac{1}{r_3} = \frac{p'}{p'''r}$.

But since the terminal pressure in the low-pressure cylinder will be that due to an initial pressure, p', and a rate of expansion, r. Then

$$\frac{p'}{r} = \frac{p'''}{r_3}; \text{ or } \frac{1}{r_3} = \frac{p'}{p'''r}.$$

Therefore.

$$\left.\begin{array}{lll} \text{Cut-off in high-pressure cylinder} & & = \dfrac{R}{r} \\ \quad ,, \quad \text{mean-pressure} & ,, & = R_1 \times \dfrac{p'}{p''r} \\ \quad ,, \quad \text{low-pressure} & ,, & = \dfrac{p'}{p'''r} \end{array}\right\} \quad . \quad . \quad (3)$$

To avoid any lengthy or elaborate calculations, a result sufficiently accurate for practical purposes may be obtained by assuming a value for x, and using it only in the first step of the calculation. This value will vary from 1·0 to 0·9 in well-proportioned engines of this class, when the steam pressure is not less than 120 lbs. absolute, and the rate of expansion not less than 10 times.

Example.—To find the mean pressures in the three-cylinder continuous expansion engine, using steam of 120 lbs. absolute pressure, and expanding 12 times. The ratio of low-pressure to high-pressure cylinder being 6, and of low-pressure to mean-pressure cylinder, 2·5; the back pressure in low-pressure cylinder being 4 lbs.:

Assume $x = 0·9$.

$$\left.\begin{array}{l} P_m = 120 \times 0·2904 = 34·85 \text{ lbs.} \\ p'_m = 120 \times 0·8465 = 101·58 \text{ lbs.} \end{array}\right\} \text{Table xvii.}$$

$$p'_m - p'' = \tfrac{6}{3}(34·85 - 4) \times 0·9 = 55·53 \text{ lbs.}$$

Therefore,

$$p'' = 101·58 - 55·53 = 46·05 \text{ lbs.}$$

Now,

$$\frac{1}{r_2} = 2·5 \times \frac{120}{46·05 \times 12} = ·544 \text{; or } r_2 = 1·838.$$

Then,

$$p''_m = p'' \frac{1 + \text{hyp. log. } 1·838}{1·838} = 38 \text{ lbs.}$$

If the work performed in the second cylinder is to equal that done in the first, then

$$p''_m - p''' = \frac{R_1}{R} \times 55·53 = \frac{2·5}{6} \times 55·53 = 23·14 \text{ lbs.}$$

Then,

$$p''' = 38 - 23·14 = 14·86 \text{ lbs.}$$

$$\frac{1}{r_3} = \frac{120}{14·86 \times 12}\text{; or } r_3 = 1·486.$$

$$p'''_m = p''' \frac{1 + \text{hyp. log. } 1·486}{1·486} = 13·96 \text{ lbs.}$$

$$p'''_m - p_0 = 13·96 - 4 = 9·96 \text{ lbs.}$$

Therefore, the mean pressures are 55·53 lbs., 23·14 lbs., and 9·96 lbs.

Referred to the low-pressure cylinder,

$$P'_m - p_0 = 9·96 + \frac{27·14}{2·5} + \frac{55·53}{6} = 28·471,$$

$$P_m - p_0 = 30·85.$$

Therefore,

$$x = \frac{28 \cdot 471}{30 \cdot 85} = 0 \cdot 923.$$

So that if the work is exactly equally divided between the cylinders, then—
Mean pressure in low-pressure cylinder

$$= \frac{P_m - p_0}{3} 0 \cdot 923 = \frac{30 \cdot 85}{3} \times 0 \cdot 923 = 9 \cdot 49 \text{ lbs.}$$

Mean pressure in mean pressure cylinder

$$= \frac{R_1}{3}(P_m - p_0)\, 0 \cdot 923 = \frac{2 \cdot 5}{3} \times 30 \cdot 85 \times 0 \cdot 923 = 23 \cdot 72 \text{ lbs.}$$

Mean pressure in high-pressure cylinder

$$= \frac{R}{3}(P_m - p_0)\, 0 \cdot 923 = \tfrac{6}{3} \times 30 \cdot 85 \times 0 \cdot 923 = 56 \cdot 94 \text{ lbs.}$$

Actual Mean Pressure in Practice.—In the preceding pages, the mean pressure spoken of is only such as would be obtained from a *perfect* engine in which steam is dry and expanding at a constant temperature, and as such is what may be called the *theoretical mean pressure.* In an actual engine, however carefully designed, manufactured, and worked, there will be certain causes of loss of pressure, so that the actual indicator-diagram will show a mean pressure considerably less than that due to the initial pressure and the rate of expansion, allowing that during expansion work has been done.

The following are the principal causes of loss of pressure in the cylinder of a marine engine :—

(1) **Frictional Resistance in the Stop-valves on the Boiler and Engine, and in the Pipes connecting these.**—If the initial pressure is taken as that in the valve case, of course this particular loss does not affect the indicator-diagram at all. If the stop-valves are opened to the extent of one-quarter of their diameter, and the steam-pipe is fairly straight and short, and of sufficient diameter, so that the flow of steam at any point does not exceed a velocity of 8,000 feet per minute : the loss of pressure at the valve-case will be very slight, and not exceed $2\frac{1}{2}$ per cent. If the capacity of the valve-case is nearly equal to that portion of the cylinder filled at the cut-off point, the loss will be still less, as the case then acts as a reservoir in which steam is stored between the cut-off and admission periods.

(2) **Friction or Wire-drawing of the Steam during admission and cut-off.**—This is one principal cause of loss of pressure in most marine engines, and is generally due to defective motion of the valve-gear, combined with small steam ports and passages. If the opening to steam during admission is small at the most, and the valve closes slowly, large passages and ports are of no avail ; and, on the other hand, if the passages are too contracted, there will be considerable loss of pressure in the cylinder, however efficient the valve-gearing may be. But the slow and limited motion of the ordinary slide-valve itself is the most serious obstacle to the obtaining of good diagrams. The slow opening of the valve causes no loss, as the piston speed is low at that period. A perfect valve should open wide enough to allow the steam to pass at a velocity of 8,000 feet per minute, and remain open until cut-off, which should take place quickly ; the valve should remain closed until very nearly the end of the stroke, when it should open quickly and wide to *exhaust ;*

the slow closing to *exhaust*, and slow opening to *lead*, are of no consequence, and cause no practical loss. The loss of pressure from these causes with engines having common slide-valves, and the ordinary link-motion, is considerable; especially is this the case when steam is cut-off early in the stroke, by setting the main valves with very little lead, and having only single ports. As has already been stated, however, the steam becomes superheated by the friction; it is, therefore, a little more efficient during expansion than it would otherwise be.

When cut-off is effected by means of special valve-gear, or by a separate cut-off valve, the pressure at cut-off is very little below that in the valve-casing, and sometimes very nearly equal to it; when effected by the ordinary single-ported slide-valve and link motion, the pressure at cut-off is sometimes as much as 15 per cent., and seldom less than 7½ per cent. below that in the valve-case.

(3) **Liquefaction during Expansion,** due partly to the cooling action of the walls of the cylinder and the passages, is a frequent source of loss of pressure, and this was especially so in expansive engines working with moist steam in unjacketed cylinders; it is observable also, though in a smaller degree, in all compound engines working under similar circumstances.

(4) **Exhausting before the Piston has reached the end of its stroke,** although conducive to the good working of a fast-running engine, will show a loss of pressure in the indicator-diagram. The loss from this cause is, however, more imaginary than real; but it must not be forgotten that the I.H.P. will be thereby less, which is important when the I.H.P. is deemed essential in the contract.

(5) **Compression and Back Pressure due to "Lead"** also tend to reduce the mean pressure of the diagram when compared with the *theoretical mean pressure*. But these are both essential to the good working of an engine, and (as has been shown in a previous part of this chapter) compression tends to balance the loss due to clearance.

(6) **Friction in the Ports, Passages, and Pipes,** between cylinder and cylinder and condenser, produces a loss of pressure, and, although not large when the velocity through them does not exceed 6,000 feet per minute, sometimes amounts to 2 or 3 lbs. in badly designed engines.

(7) **Clearance** has been shown to serve to increase the mean pressure beyond that due to the nominal rate of expansion, and therefore cannot be reckoned as a source of loss, unless the *equivalent* cut-off is taken to obtain the rate of expansion.

It will be seen, then, that the actual mean pressure expected to be deduced from the indicator-diagram of an engine depends very much on the proportion and arrangement of the cylinders and their valves, etc., and in calculating the *expected* mean pressure from the *theoretical mean pressure*, due allowance must be made in each individual case.

If the theoretical mean pressures be calculated by the methods laid down in this chapter, and the necessary corrections made for *clearance* and compression, the *expected* mean pressure may be found by multiplying the results by the factor in the following Table xviii.

If no correction be made for the effects of clearance and compression, and the engine is in accordance with general modern practice, the clearance and compression being proportionate, then the Theoretical Mean Pressure may be multiplied by 0·96, and the product again multiplied by the proper factor in Table xviii., the result being the expected mean pressure.

TABLE XVIII.

Particulars of Engine.	Factor.
(1) Expansive engine, special valve-gear, or with a separate cut-off valve, cylinders jacketed,	0·94
(2) Expansive engine having large ports, etc., and good ordinary valves, cylinders jacketed,	0·9 to 0·92
(3) Compound engines, with ordinary slide valves, cylinders jacketed, and good ports, etc.,	0·8 to 0·85
(4) Compound engines as in general practice in the merchant service, with early cut-off in both cylinders, without jackets and expansion valves,	0·7 to 0·8
(5) Triple- and quadruple-compound engines, with ordinary slide-valves, good ports, unjacketed, moderate piston speed,	0·65 to 0·75
(6) Fast-running engines of the type and design usually fitted in war-ships, and express with fast-running engines,	0·6 to 0·7

Example.—To find the expected mean pressure in the cylinders of a marine engine using steam of 60 lbs. absolute pressure, the rate of expansion 4. the clearance equal to one-tenth of the cylinder, and the pressure in the condenser 2 lbs., the valve-gearing specially adapted for an early cut-off, and the ports, passages, etc., of ample size; compression commences at $\frac{3}{4}$ of the stroke. The cylinders are jacketed.

The effective rate of expansion is

$$r_1 = 4 \times \frac{1 + 0{\cdot}1}{1 + 4 \times 0{\cdot}1} = 3{\cdot}143.$$

$$p^e = \frac{0{\cdot}25 + 0{\cdot}1}{0{\cdot}1} \times 2 = 7 \text{ lbs.}$$

and the rate of expansion 3·5. Then,

$$p^0_m = 7 \times \frac{1 + \text{hyp. log. } 3{\cdot}5}{3{\cdot}5} = 4{\cdot}5 \text{ lbs.}$$

$$p'_m = 60 \times \frac{1 + \text{hyp. log. } 3{\cdot}143}{3{\cdot}143} = 41 \text{ lbs.}$$

$$\text{Expected mean pressure} = 41\,(1 + 0{\cdot}1) - 60 \times 0{\cdot}1 - 2\,(1 + 0{\cdot}5 - 0{\cdot}1) - 4{\cdot}5\,({\cdot}25 + 0{\cdot}1) = 35{\cdot}3 \text{ lbs.}$$

If the effects of clearance and cushioning be neglected, the mean pressure $= 60 \times 0{\cdot}5965 - 2$, or 33·8 lbs. This is less than the result obtained by the more accurate calculation in this case, because the cushioning is small for so low a back pressure when compared with the clearance.

The mean pressure in practice will be found now by multiplying 35·3 lbs. by 0·94, and is therefore 33·18 lbs.

Example.—To find the expected mean pressure in the cylinders of a compound engine using steam of 100 lbs. absolute pressure, the cut-off in both high-pressure and low-pressure cylinders being at half stroke; the clearance in both cylinders is equal to one-tenth of their net capacity; the pressure in the condenser is 2 lbs.; the cylinders are jacketed, and the ports, etc., of ample size, no expansion valves. Compression commences at $\frac{3}{4}$ the stroke. Cylinder ratio 4.

Here the effective rate of expansion $= 2\dfrac{1 + 0{\cdot}1}{1 + 2 \times 0{\cdot}1} = 1{\cdot}83$

The theoretical pressure in the receiver $= \dfrac{100}{1{\cdot}82} \times \dfrac{2}{4} = 27{\cdot}3$ lbs.

The *expected* pressure in receiver $= 27{\cdot}3 \times 0{\cdot}85 = 23{\cdot}2$ lbs.

The steam is compressed in high-pressure cylinder to

$$p^c = \frac{\frac{1}{4} + \frac{1}{10}}{\frac{1}{10}} \times 23{\cdot}2 = 81{\cdot}2 \text{ lbs.}$$

The rate of compression $= \dfrac{\frac{1}{4} + \frac{1}{10}}{\frac{1}{10}} = 3{\cdot}5.$

The mean pressure due to a rate of expansion 1·83, and an initial pressure of 100 lbs.

$$= 100\,\frac{1 + \text{hyp. log. } 1{\cdot}83}{1{\cdot}83} = 87 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·5, and an initial pressure of 81·2 lbs.

$$= 81{\cdot}2\,\frac{1 + \text{hyp. log. } 3{\cdot}5}{3{\cdot}5} = 52 \text{ lbs.}$$

The theoretical mean pressure in high-pressure cylinder

$$= 87\,(1 + 0{\cdot}1) - 100 \times 0{\cdot}1 - 23{\cdot}2\,(1 - 0{\cdot}5 - 0{\cdot}1) - 52\,(0{\cdot}25 + 0{\cdot}1)$$
$$= 58{\cdot}22 \text{ lbs.}$$

The *expected mean pressure* in high-pressure cylinder

$$= 58{\cdot}22 \times 0{\cdot}85 = 49{\cdot}5 \text{ lbs.}$$

The mean pressure due to a rate of expansion 1·83, and an initial pressure of 23·2 lbs.

$$= 23{\cdot}2\,\frac{1 + \text{hyp. log. } 1{\cdot}83}{1{\cdot}83} = 20{\cdot}2 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·5, and an initial pressure of 7 lbs.

$$= 4{\cdot}5 \text{ lbs.}$$

Then theoretical mean pressure in low-pressure cylinder

$$= 20{\cdot}2\,(1 + 0{\cdot}1) - 23{\cdot}2 \times 0{\cdot}1 - 2\,(1 - 0{\cdot}5 - 0{\cdot}1) - 4{\cdot}5\,(0{\cdot}25 + 0{\cdot}1)$$
$$= 17{\cdot}5 \text{ lbs.}$$

And the *expected mean pressure* in low-pressure cylinder

$$= 17{\cdot}5 \times 0{\cdot}85 = 14{\cdot}87 \text{ lbs.}$$

Example.—To find the expected mean pressure in a compound engine as fitted formerly in merchant steamers; the cylinders are unjacketed, the boiler pressure 80 lbs. (95 lbs. absolute); the cylinder ratio is 3·5, and the cut-off, effected by common slide-valves, is at half-stroke in the high-pressure cylinder, and 0·6 the stroke in low-pressure cylinder. The clearance in both cylinders is one-twelfth the cylinder capacity. Compression takes place in the high-pressure cylinder when the piston is 0·2 of its stroke from the end, and in the low-pressure cylinder at 0·3.

Efficiency in this case taken at 0·75.

The effective rate of expansion in high-pressure cylinder

$$= 2\frac{1 + \frac{1}{12}}{1 + \frac{2}{12}} = 1{\cdot}86.$$

The theoretical pressure in the receiver

$$= \frac{95}{1{\cdot}86} \times \frac{1}{3{\cdot}5 \times 0{\cdot}6} = 24{\cdot}3 \text{ lbs}$$

The expected pressure in the receiver

$$= 24{\cdot}3 \times 0{\cdot}75 = 18{\cdot}23.$$

The rate of compression in the high-pressure cylinder

$$= \frac{0{\cdot}2 + 0{\cdot}083}{0{\cdot}083} = 3{\cdot}4,$$

and the steam is compressed to 18·23 × 3·4, or 62 lbs. The mean pressure due to a rate of expansion of 1·86, and an initial pressure of 95 lbs.

$$= 95\,\frac{1 + \text{hyp. log. } 1{\cdot}86}{1{\cdot}86} = 80 \text{ lbs.}$$

The mean pressure due to a rate of expansion 3·4, and an initial pressure of 62 lbs.

$$= 62\,\frac{1 + \text{hyp. log. } 3{\cdot}4}{3{\cdot}4} = 40 \text{ lbs.}$$

Then theoretical mean pressure in high-pressure cylinder

$$= 80(1 + \tfrac{1}{12}) - 95 \times \tfrac{1}{12} - 18{\cdot}23(1 - 0{\cdot}4 - \tfrac{1}{12}) - 40(0{\cdot}2 + \tfrac{1}{12})$$
$$= 58 \text{ lbs.}$$

And the expected mean pressure in high-pressure cylinder

$$= 58 \times 0{\cdot}75 = 43{\cdot}5 \text{ lbs.}$$

The back pressure in the condenser is 2 lbs.

The effective rate of expansion in low-pressure cylinder

$$= \frac{1}{0{\cdot}6} \times \frac{1 + \dfrac{1}{12}}{1 + \dfrac{1}{0{\cdot}6}\;\dfrac{1}{12}} = 1{\cdot}58.$$

The rate of compression in low-pressure cylinder

$$= \frac{0{\cdot}3 + 0{\cdot}083}{0{\cdot}083} = 4{\cdot}6.$$

Steam is compressed in low-pressure cylinder to 4·6 × 2, or 9·2 lbs.

The mean pressure due to a rate of expansion 1·58, and an initial pressure of 18·23 lbs.

$$= 18{\cdot}23 \times \frac{1 + \text{hyp. log. } 1{\cdot}58}{1{\cdot}58} = 16{\cdot}8 \text{ lbs.}$$

The mean pressure due to a rate of expansion 4·6, and an initial pressure 9·2 lbs.

$$= 9{\cdot}2 \times \frac{1 + \text{hyp. log. } 4{\cdot}6}{4{\cdot}6} = 5 \text{ lbs.}$$

Then theoretical mean pressure in low-pressure cylinder

$$= 16{\cdot}8\,(1 + \tfrac{1}{12}) - 18{\cdot}23 \times \tfrac{1}{12} - 2\,(1 - 0{\cdot}6 - \tfrac{1}{12}) - 5\,(0{\cdot}3 + \tfrac{1}{12})$$
$$= 14{\cdot}13 \text{ lbs.}$$

And the *expected mean pressure* in low-pressure cylinder

$$= 14{\cdot}13 \times 0{\cdot}75 = 10{\cdot}6 \text{ lbs.}$$

A Practical Method of Estimating the Expected Mean Pressure in a compound engine may be followed by first calculating the theoretical mean pressure due to the total rate of expansion, subtracting from it the back pressure in condenser, and dividing by 2 for a two-cylinder engine, by 3 for a triple, and by 4 for a quadruple engine. The result is the mean pressure in the low-pressure cylinder, when there is no loss from "drop." Multiply this by the factor in Table xviii., and again multiply the product by 0·8, and the *result is the expected mean pressure when the work is equally divided between the two cylinders.*

Example.—To find the mean pressures expected in the cylinders of a compound engine, using steam of 90 lbs. absolute pressure, and expanding it six times; the ports being of ample size, and the cylinders jacketed; the cut-off in high-pressure cylinder effected by an expansion-valve, and the pressure in the condenser is 3.

The mean pressure due to a rate of expansion of 6, and initial pressure of 90 lbs., ... $= 90 \times \cdot 4653 = 41 \cdot 8$ lbs.
The effective mean pressure . . . . $= 41 \cdot 8 - 3 = 38 \cdot 8$ lbs.
The effective mean pressure in L.P. cylinder $= 19 \cdot 4$ lbs.
The expected mean pressure in L.P. cylinder $= 19 \cdot 4 \times 0 \cdot 9 \times 0 \cdot 8 = 13 \cdot 9$ lbs.

If the ratio of the cylinder is 3·5, then

The expected mean pressure in H.P. cylinder $= 13 \cdot 9 \times 3 \cdot 5 = 48 \cdot 7$ lbs.

Graphic Method.—It has been assumed in this chapter that steam expands in accordance with Boyle and Marriott's law—viz., $p\,v =$ *constant*—on account of its convenient form and easy calculation. As a matter of fact, the results obtained in this way are sufficiently accurate for practical purposes, although, of course, they are not scientifically accurate, for steam does not expand in practice at a constant temperature as is assumed in Boyle's law, and, moreover, when expanding in practice, it is doing work, and therefore giving up some of its heat with a corresponding reduction in pressure.

The simpler and more easily-worked methods of obtaining the division of power in compound engines are graphic—*i.e.*, by means of geometrical diagrams based on the same physical facts as before.

Draw a straight line A C (fig. 55) to represent the capacity of the low-pressure cylinder, including its clearance, which latter is represented by A B.

Draw an ordinate, A K, representing the absolute pressure of the steam.

Take a point, D, on A C, so that $\frac{A\,C}{A\,D}$ is the total rate of expansion required in the system.

Then draw the expansion curve D Q′ C′.

Take a point F′ in K D, so that K F′ represents the clearance of the high-pressure cylinder.

Draw F′ F parallel to A K.

Now, K D′ represents the capacity of the high-pressure cylinder, including its clearance at the point of cut-off in that cylinder.

Then, take a point L in A C, so that $\frac{F\,D}{F\,L}$ = rate of cut-off in the high-pressure cylinder, and draw L L′ parallel to A K.

A L will, therefore, represent the capacity of the high-pressure cylinder, including its clearance A F.

Now, take point N on A C, so that A N represents the clearance in the medium-pressure cylinder.

Take a point, V, in A C, and a point, Q, in A C, so that A Q represents the volume of the medium-pressure cylinder, including its clearance, A N, and $\frac{\text{N V}}{\text{N Q}}$ = the rate of cut-off in the medium-pressure cylinder.

Draw V V′ and Q Q′ parallel to A K.

Take a point, S, in A C, so that $\frac{\text{B S}}{\text{B C}}$ = the rate of cut-off in low-pressure cylinder.

Draw S S′ parallel to A K, and S′ R parallel to A C.

Then F′ D′ L′ L″ F″ is the theoretical high-pressure diagram.

N′ V′ Q′ Q″ N″ is the medium-pressure diagram, and B′ S′ C′ C B the low-pressure diagram.

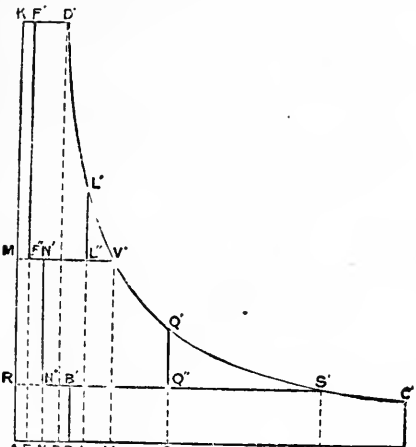


Fig. 55.—Theoretical Diagram for Triple-Expansion.

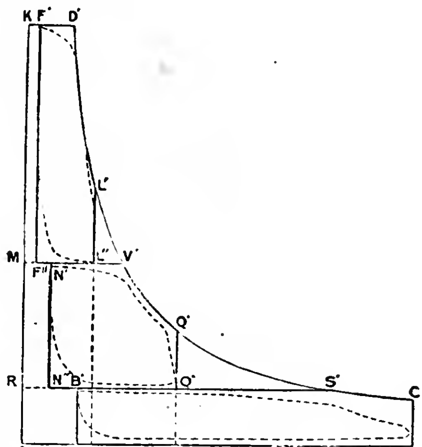


Fig. 55*a*.—Diagrams as in Practice, from Theoretical Expansion Diagram.

In actual practice the indicator diagrams differ from the theoretical ones, for reasons already given.* They may, however, be inscribed within the theoretical diagram of the mean pressure, as shown in fig. 55*a*, which has been drawn in accordance with the method prescribed above, so that F′ D′ F″ is the high-pressure diagram, V Q N the medium, and S C′ C B the low. In practice, the area of the actual diagram of the high pressure is 77 to 80 per cent. of the theoretical. The actual medium-pressure diagram is 70 to 73 per cent. of the theoretical, and the low pressure 55 to 60 per cent.

In fast-running engines the percentage is, of course, less than that obtained from the diagrams of slow-moving engines.

If it is found by this means that the power is not sufficiently evenly divided, the cuts-off in the medium-pressure and low-pressure cylinders must be modified. For example, if the power in the low-pressure cylinder is too small, the cut-off point, S, must be moved nearer to A, so that the figure, B′ S′ C′ C B, is enlarged at the expense of the medium-pressure diagram, N′ V′ Q′ Q″ N″.

* For a refinement in method of construction, *vide* Appendix.

TABLE XIX.—RESULTS OF TRIALS OF ENGINES OF NAVAL SHIPS (TWIN SCREW)

Name.	Nature of Trials.	Cylinders. Diameters. H.P.	M.P.	L.P.	L.P.	Stroke. Inches.	Engine Room. Steam. lbs.	Vacuum. Inches.	Revolutions.	Cut-Off. H.P.	Mean Pressures. H.P.	M.P.	L.P.	L.P.	Referred to L.P. Cyl.	One Engine. Indicated Horse Power. H.P.	M.P.	L.P.	L.P.	Total.	Coal per I.H.P.	Nominal Rate of Expansion.
Good Hope,	Full S. Red. S.	43½	71	81½	81½	48	278 268	27 26·5	126 110·7	0·73 0·63	105 95·8	44·6 37·9	16·9 11·9	17·9 13 4	49·3 40·7	4,764 3,819	6,399 4,029	2,698 1,606	2,849 1,864	15,710 11,378	1·70 1·75	9·60 ..
Argonaut,	Full S. Red S.	34	55½	64	64	48	236 242	25·5 27	129·8 115·7	0·71 0·60	96·5 89	32·2 27·9	19·2 17·6	23·1 21·5	46·1 42·7	2,704 2,266	2,406 1,900	1,908 1,595	2,295 1,950	9,313 7,711	1·66 1·73	9·99 11·82
Formidable,	Full S. Red. S.	31½	51½	84	..	51	245 240	25·5 26·5	110·2 101	0·70 ..	115 95·2	44 34·4	17·9 14·3	.. ..	50 5 40·7	2,543 1,934	2,601 1,868	2,815 2,066	.. ..	7,959 5,868	1·80 2·00	10·10 ..
St. George,	Full S.	40	59	88	..	51	154	27·8	100·1	0·67	45·2	24·3	13·1	..	33·4	1,465	1,713	2,055	..	5,233	1·62	7·22
Friesland,	Full S.	33	49	74	..	39	200	24·8	148	..	62	36·5	14·74	..	43·2	1,549	2,001	1,851	..	5,401	..	..
Arrogant,	Full S. Red. S.	26	42	68	..	39	245 235	26 27	143 128	0·70 0·514	112·5 86·5	47·1 35·3	20·5 15·8	.. ..	52·8 41·2	1,685 1,159	1,841 1,233	2,100 1,449	.. ..	5,406 3,841	.. ..	9·77 13·50
Antrim,	Full S. Red. S.	41½	65½	73½	73½	42	185 161	25·0 25·5	142·6 87·7	.. ..	82·6 38·12	32·5 12·05	16·7 3·77	16·6 3·84	42·7 14·7	3,383 960	3,315 756	2,143 299	2,136 303	10,977 2,318	2·26 2·06	.. ..
Pearl,	Force D. Natl. D.	30½	45	68	..	33	145 149	24·8 27	158·3 143·1	0·65 ..	56·9 46·3	30 23·7	13·1 10·9	.. ..	37·1 30·6	1,098 806	1,260 901	1,253 942	.. ..	3,611 2,649	.. ..	5·85 ..
Perseus,	Full S. Red. S.	20½	33	54	..	27	266 251	24·2 25·1	214·8 197	0·75 ..	.. ..	.. ..	.. ..	.. ..	52·5 42·9	992 707	1,263 987	1,268 942	.. ..	3,523 2,636	.. 2·27	9·33 ..
Forward,	Full S. Red. S.	31½	50½	57	57	30	227 50	25·3 21·0	210·6 80·3	0·58 0·44	109 17·1	34·5 5·7	14·85 1·87	14·83 1·76	44·5 6·6	2,727 162	2,224 139	1,218 68	1,217 57	7,386 414	2·71 2·66	11·4 15·0
Topaze,	Full S. Red. S.	24¼	38½	42½	42½	24	271 198	24·0 25·0	245·5 150·3	.. ..	111·0 34·63	47·2 15·15	21·5 10·36	22·7 9·75	59 5 22·9	1,534 292	1,637 321	897 265	946 249	5,014 1,127	2·65 2·06	.. ..
Dove,	Force D	19½	30	34	34	18	206	25	385·4	0·75	77·3	45·5	17·7	17·2	47·4	808	1,127	563	547	3,045	2·38	8·09
Snapper,	Force D.	19¼	28½	43	..	18	180	25	370·6	0·58	64·5	28·7	14·6	..	39·7	635	592	715	..	1,942	..	8·62
Speedy,	Force D. Natl. D.	22	34	51	..	21	198 189	27·5 27·5	247 209·7	.. ..	75·6 61·5	33·4 26·3	15·6 12·1	.. ..	44·5 35·4	749 522	795 533	847 551	.. ..	2,391 1,606	.. ..	.. ..

TABLE XX.—Results of Trials of Engines of Merchant Ships.

Name.	Twin or Single Screw.	Cylinders.					Engine Room.									One Engine.					Nominal Rate of Expansion.
		Diameters.				Stroke. Inches.	Steam. Lbs.	Vacuum. Inches.	Revolutions.	Cut-Off. H.P.	Mean Pressures.					Indicated Horse-power.					
		H.P.	M.P.	3rd Cylr.	L.P.						H.P.	M.P.	3rd Cylr.	L.P.	Referred to L.P.	H.P.	M.P.	3rd Cylr.	L.P.	Total.	
Auguste Victoria,	T.S.	41	67	106	..	63	149·3	24	78	0·78	62·6	27·8	..	11·16	32·8	2,214	2,470	..	2,494	7,178	8·58
Kherson,	T.S.	36	57	92	..	54	155	27¼	90·6	..	82·9	36·8	..	14·2	41·0	2,085	2,320	..	2,335	6,740	..
Saxonia,	T.S.	29	41½	59	84	54	192	25·3	78	..	73·1	41	20·9	12·08	39·0	985	1,158	1,202	1,420	4,765	..
Liguria,	S.S.	33	54	88	..	60	183	25	72	..	77·5	30	..	12	34·2	1,444	1,499	..	1,502	4,535	..
Cazengo,	S.S.	32	48	80	..	48	145	..	71	0·656	68·1	32·0	..	11·0	33·6	943	994	..	947	2,884	9·50
Colorado,	S.S.	31	50	82	..	57	160	27½	66	0·60	60·8	23·1	..	11·06	28·3	871	862	..	1,109	2,842	11·67
Othello,	S.S.	24¾	40	70	..	48	200	28	71	0·59	88·5	36·8	..	13·6	36·7	733	794	..	903	2,430	15·25
Ariosto	S.S.	30	46	75	..	45	148	..	79	0·63	57·4	33·4	..	11·05	32·8	728	996	..	876	2,600	9·92
Eldorado.	S.S.	28	43	70	..	39	154	28	88	0·60	57·6	31·2	..	10·7	31·6	616	785	..	710	2,112	10·40
Amsterdam,	T.S.	26	39½	61	..	36	155	26¼	135·6	0·71	62·1	28·4	..	14·5	37·8	812	857	..	1,048	2,717	7·75
P. George.	T.S.	26	40	45	45	30	177	27	177	0·39	57·5	50·3	15·4	15·3	36·9	818	1,019	682	630	3,149	8·89
Ohio,	S.S.	22½	32	45	61	42	200	27½	67·5	0·64	63·5	28·5	20·4	11	39·0	362	328	464	507	1,661	12·66
P. Edward,	T.S.	19	30	48	..	24	180	25	192·3	0·68	81	33·5	..	13·7	39·6	536	554	..	577	1,667	9·37
Castro,	S.S.	19	33	56	..	36	200	25	91·5	0·59	87·8	33·5	..	10·9	32·5	417	470	..	445	1,333	14·69
Iona,	S.S.	22	34	57	..	39	150	27·8	61·5	0·25	47	20	..	7·2	21·7	208	217	..	220	645	26·88
Colchester,	T.S.	14½	23	27	27	33	180	26·5	145	0·75	83	38·0	18·4	18·2	44·6	331	382	255	252	1,220	9·33
Bolma,	S.S.	18	27	48	..	24	181	26·7	89	0·56	75	41·3	..	10·25	38·1	231	286	..	225	742	12·71
Righi,	S.S.	17	25	35	50	27	184	25	72	0·57	61·3	32·6	20·1	7·8	32·7	136	157	189	151	633	15·18
Volopo,	S.S.	13	22	36	..	24	200	22·6	112	0·60	96·7	38·7	..	13·1	40·0	174	199	..	181	554	12·75
Polo,	S.S.	12¾	18½	27	39	24	200	25	98	0·62	75	34·3	15·3	11·85	34·8	113	110	104	168	495	15·05
Gertie,	S.S.	12	20	32	..	23	200	22	136	0·58	74·5	38·8	..	15·5	41·0	133	192	..	197	522	12·26
Vasari.	S.S.	26½	37½	59	77	57	230	27½	75	..	80·6	44·7	18·1	10·6	41·3	964	1,068	1,059	1,068	4,159	..
Kingswear,	S.S.	19	31	51	..	36	180	27¼	78	..	80	34	..	13·58	37·2	322	364	..	394	1,080	..
Patagon,	S.S.	24	34½	49	71	48	220	28	65	..	80·3	32·1	16·0	9·12	33·5	572	476	472	568	2,088	..
Lisboa,	S.S.	22	35	53	..	33	205	26	114	..	84·2	33·2	..	9·5	31·55	350	400	..	330	1,080	..
R.W.,	S.S.	25	40	68	..	45	180	27½	61	..	68·7	27·1	..	10·15	30·6	500	502	..	545	1,547	..
Arctic,	S.S.	12	18	30	..	18	155	28	136	0·62	67·8	29·1	..	11·4	..	92·1	91·4	..	99·5	283	10·1

TABLE XXI.—Results of Trials of Two-Stage Compound Marine Engines.

General Description of Engines.	Cylinders.		N.H.P.	Boiler Pressure.	Vacuum.	Revolutions.	Rate of Total Expansion.	Cylinder Mean Pressure.		I.H.P.		
	Diameter.	Stroke.						H.P.	L.P.	H.P.	L.P.	Total.
	Ins.	Ins.		Lbs.	Ins.			Lbs.	lbs.			
Screw Vertical Compound,	1 of 62 2 of 90	66	1,000	86	27	55	..	44	16·5 16·7	2,433	1,945 1,928	6,306
Do. do.,	43 and 86	60	420	80	27	57·4	7·0	47·4	11·2	1,198	1,131	2,329
Do. do.,	46 and 87	57	400	76	28	57	5·84	41·2	12·7	1,124	1,238	2.362
Do. do.,	42 and 84	60	400	88	$26\frac{3}{4}$	57	7·2	51·0	11·7	1,220	1,127	2,347
Do. do.,	38 and 76	48	270	79	$26\frac{1}{2}$	62	6·62	46·4	11·2	790	763	1,553
Do. do.,	40 and 74	48	250	80	27	63	5·67	41·1	12·26	789	805	1,594
Do. do.,	32 and 64	63	250	80	25	53	6·9	49·7	11·67	674	633	1,307
Do. do.,	36 and 72	48	230	72	$27\frac{1}{4}$	60	6·0	45·2	10·3	669	610	1,279
Do. do.,	35 and 69	39	180	82	$26\frac{3}{4}$	70	6·6	46·4	11·5	614	593	1,207
Do. do.,	$33\frac{3}{4}$ and $64\frac{1}{2}$	42	180	70	28	58	5·55	40·3	11·3	442	454	896
Do. do.,	32 and 62	36	150	66	25	69	6·43	45·4	10·37	458	392	851
Do. do.,	25 and 50	45	130	80	27	71	7·5	46·2	12·5	366	397	763
Do. do.,	30 and 55	36	125	84	$24\frac{1}{2}$	74·5	6·96	45·1	11·9	430	383	813
Do. do.,	27 and 53	33	110	80	24	84	7·49	46·1	11·75	369	361	731
Do. do.,	27 and 50	30	100	73	$23\frac{1}{2}$	68	5·42	43·6	13·36	260	270	530
Do. do.,	24 and 45	33	90	84	26	82	6·43	45·1	13·0	278	282	560
Do. do.,	21 and 40	27	60	77	$25\frac{1}{2}$	98	6·15	45·1	11·37	209	191	400
Do. do.,	17 and 32	18	38	90	$27\frac{3}{4}$	142	5·55	49·5	11·3	168·5	136·4	305
Do. do.,	30 and 57	36	137	86	$26\frac{1}{2}$	89	5·65	51·8	14·45	595	588	1,183
Do. do.,	34 and 64	39	166	86	26	75	6·00	45·6	11·93	612	567	1,179
Do. do.,	26 and 56	60	224	115	$27\frac{1}{2}$	57·5	7·73	66·9	14·5	619	622	1,241
Do. do.,	12 and 22	20	30	85	26	160	5·60	44·3	13·8	80·8	84·8	165·6

TABLE XXII.—Results of Trials of Triple-Expansion Three-Crank Engines.

Ship.	Screws, Number of.	Cylinders. Diameters. H.P.	Cylinders. Diameters. M.P.	Cylinders. Diameters. L.P.	Cylinders. Stroke.	Engine Room. Steam.	Engine Room. Vacuum.	Engine Room. Revolutions.	Engine Room. Cut-off in H.P. Cylr.	Engine Room. Mean Pressures. H.P.	Engine Room. Mean Pressures. M.P.	Engine Room. Mean Pressures. L.P.	Engine Room. Mean Pressures. Referred to L.P. Cylr.	One Engine Only. Indicated Horse-power. H.P.	One Engine Only. Indicated Horse-power. M.P.	One Engine Only. Indicated Horse-power. L.P.	One Engine Only. Indicated Horse-power. Total.	Nominal Rate of Expansion.
		Ins.	Ins.	Ins.	Ins.	Lbs.	Ins.			Lbs.	Lbs.	Lbs.	Lbs.					
S.S. Ag. V., . .	Twin	41	67	106	63	149	24·0	78·0	0·78	62·6	27·8	11·16	32·8	2,214	2,470	2,494	7,178	8·58
S.S. Krsn., . .	do.	36	57	92	54	155	27·3	90·6	..	82·9	36·8	14·2	41·0	2,085	2,320	2,335	6,740	..
S.S. Lgra., . .	do.	33	54	88	60	183	25	72·0	..	77·5	30·0	12·0	34·2	1,444	1,490	1,592	4,535	..
H.M.S. Ldn., .	do.	31½	51½	84	51	245	25·5	110	..	115·0	44·0	17·9	50·0	2,543	2,601	2,815	7,859	..
H.M.S. Argt., .	do.	26	42	68	39	245	26·0	143	0·70	112·5	47·1	20·5	52·8	1,685	1,841	2,100	5,406	9·80
S.S. Czgo., . .	do.	32	48	80	48	145	26·0	71·0	0·66	68·1	32·0	11·0	33·6	943	994	947	2,884	9·50
S.S. Otlo., . .	do.	24¾	40	70	48	200	28·0	71·0	0·59	88·5	36·8	13·6	36·7	733	794	903	2,430	15·25
S.S. R.W., . .	Single	25	40	68	45	180	27·5	61·0	..	68·7	27·1	10·15	30·6	500	502	545	1,547	..
S.S. Vna., . .	Twin	26	39½	61	36	155	26·5	135	0·72	62·4	31·2	14·15	38·0	810	933	1,011	2,754	7·80
S.S. Ped., . .	do.	19	30	48	24	180	25·0	192	0·68	81·0	33·5	13·7	39·6	536	554	577	1,667	9·37
S.S. Cstr., . .	Single	19	33	56	36	200	25·0	91·5	0·59	87·8	33·5	10·9	32·5	417	470	445	1,333	14·70
H.M.S. Supr., .	Twin	19¼	28½	43	18	180	25·0	371	0·58	64·5	28·8	14·6	39·5	635	592	715	1,942	8·62
H.M.S. Chgr., .	do.	18	26	39½	18	160	25·0	363	0·81	62·0	37·8	13·5	41·9	523	626	547	1,696	6·00
S.S. Mlpo., . .	Single	13	22	36	24	200	23·5	112	0·60	96·7	38·7	13·12	40·0	174	199	181	554	12·75
S.S. Iona, . .	do.	22	34	57	39	150	27·8	61·5	0·25	47·0	20·0	7·2	21·7	208	217	220	645	26·88
S.S. Arte., . .	do.	12	18	30	18	155	28·0	136	0·62	67·8	29·1	11·4	32·5	92·1	91·4	99·5	283	10·1
S.S. Zde., . .	Twin	12½	19	32	24	175	27·0	148	0·68	69·5	36·5	13·75	36·8	152	182	197	531	9·62
S.S. Gte., . .	Single	12	20	32	23	200	22·0	136	0·58	74·5	38·8	15·50	41·0	133	192	197	522	12·26

TABLE XXIII.—Results of Trials of Four-Crank Triple-Expansion Engines.

Name.	Number of Screws.	Cylinders. Diameter H.P.	Cylinders. Diameter M.P.	Cylinders. Diameter L.P.	Cylinders. Diameter L.P.	Cylinders. Stroke.	Engine Room. Steam.	Engine Room. Vacuum.	Engine Room. Revolutions.	Engine Room. Cut-off in H.P. Cylr.	Engine Room. Mean Pressures. H.P.	Engine Room. Mean Pressures. M.P.	Engine Room. Mean Pressures. L.P.	Engine Room. Mean Pressures. L.R.	Engine Room. Mean Pressures. Ref. L.P.	One Engine Only. Indicated Horse-Power. H.P.	One Engine Only. Indicated Horse-Power. M.P.	One Engine Only. Indicated Horse-Power. L.P.	One Engine Only. Indicated Horse-Power. L.P.	One Engine Only. Indicated Horse-Power. Total	Nominal Rate of Expansion.
		Ins	Ins.	Ins.	Ins	Ins.	Lbs.	Ins.			Lbs.	Lbs.	Lbs.	Lbs.	Lbs.						
H.M.S. Gh.	Twin	43½	71	81½	81½	48	278	27·0	126	..	105	44·6	16·9	17·9	50·8	4,764	5,399	2,698	2,849	15,710	..
H.M.S. Wr.;	do.	43½	69	77	77	42	194	26·7	138	..	72·6	32·7	18·5	19·2	44·1	3,147	3,568	2,513	2,607	11,834	..
H.M.S. Am.,	do.	41½	65½	73½	73½	42	185	25·0	143	..	82·6	32·5	16·7	16·6	..	3,383	3,315	2,143	2,136	10,977	..
H.M.S. Cd.,	do	37	60	69	69	42	296	26·5	145	..	105·2	41·3	19·1	18·8	49·3	3,484	3,594	2,194	2,170	11,442	..
H.M.S. Dm.,	do.	34	55½	64	64	48	245	27·0	120	..	90·0	34·0	21·9	21·7	50·3	2,370	2,386	2,044	2,025	8,825	..
H.M.S. Fd.	do.	31½	50½	57	57	30	227	25·3	212	..	109	34·5	14·9	14·8	..	2,727	2,224	1,218	1,217	7,386	..
H.M.S. Tz.,	do.	24¼	38½	42½	42½	24	271	24·0	246	..	111	47·2	21·5	22·7	57·3	1,534	1,637	897	946	5,014	..
S.S. Par,	do.	26	40	45	45	30	177	27·0	177	0·69	57·5	30·3	15·4	15·3	36·9	818	1,019	682	630	3,149	8·89
S.S. Cr.,	do.	14½	23	27	27	33	180	26·5	145	0·75	83	38·0	18·4	18·2	44·6	331	382	255	252	1,220	9·33
H.M.S. Dv.,	do.	19½	30	34	34	18	206	25·0	385	..	77·3	45·5	17·7	17·2	..	808	1,127	563	547	3,045	..
H.M.S. Sr.,	do.	18¾	27¼	28	28	18	194	24·1	391	..	63·7	33·0	19·0	18·7	..	626	684	412	406	2,128	..

TABLE XXIV.—Results of Trials of Quadruple-Expansion Engines.

Ship.	Number of Screws.	Cylinders					Engine Room.									One Engine Only.					Nominal Rate of Expansion
		Diameters.				Stroke.	Steam.	Vacuum.	Revolutions.	Cut-off in H.P. Cylinder.	Mean Pressures.					Indicated Horse-Power					
		H.P.	M.P.₁	M.P.₂	L.P.						H.P.	M.P.₁	M.P.₂	L.P.	Refd to L.P.	H P	M.P.₁	M.P.₂	L.P.	Total.	
		Ins.	Ins	Ins.	Ins.	Ins.	Lbs.	Ins.			Lbs.	Lbs.	Lbs.	Lbs.	Lbs.						
S.S. Sxa., . .	Twin	29	41½	59	84	54	192	25·3	78	..	73·1	41·0	20·9	12·08	39·0	985	1,158	1,202	1,420	4,765	..
S.S. Ptn., . .	Single	24	34½	49	71	48	220	28·0	65	..	80·3	32·1	16·0	9·12	33·5	572	476	472	568	2,088	..
S.S. Vsr, . .	do.	26½	37½	59	77	57	230	27·5	75	..	80·6	44·7	18·4	10·6 o	41·3	964	1,068	1,059	1,068	4,159	..
S.S. Oho., . .	do.	22½	32	45	64	42	200	27·5	67·5	0·64	63·5	28·5	20·4	11·0	39 0	362	328	464	507	1 661	12·66
S.S. Rgi., . .	do.	17	25	35	50	27	184	25·0	72·0	0 57	61·3	32·6	20·1	7·8	32·7	136	157	181	151	633	15·18
S.S. Plo., .	do.	12¾	18½	27	39	24	200	25·0	98	0·62	75·0	34·3	15·3	11·85	34·8	113	110	104	168	496	15·05
S.S. Cyk., . .	do.	20	28	40	57	42	160	24·8	48·5	0·60	46·1	26·9	14·3	6·98	26·1	149	169	183	181	682	13·5
S.S. Tk., . .	do	20	28	39½	57	33	212	25·0	140	0·63	73·5	37·0	19·8	10·0	40·0	540	532	565	596	2,233	12·9
S.S. Hl., . .	do.	22½	31½	45	66	51	215	25·5	76	0·57	74	35·5	18·3	8·3	33·3	570	542	566	550	2,228	15·1

CHAPTER V.

STEAM USED AFTER EXPANSION—TURBINES.

The Turbine is essentially a Velocity Machine, inasmuch as the velocity alone of the steam is the active principle, and pressure has no part whatever in producing its movement. There are rotary machines of various kinds, in which pressure does produce motion, they are, however, not turbines.

Turbines are of Two Elementary Kinds—the one works by means of the jet of steam issuing at high velocity from a generating nozzle acting impulsively on the blades or vanes of a rotor wheel secured on a shaft, as shown in fig. 51, thus causing it to turn round at a high rate of revolution. Such a machine was invented by Branca in 1630, and it is analogous in hydraulics to the undershot-mill wheel, and to the more modern and more refined Pelton wheel; since it is operated by the impulse of the steam particles on the vanes, it is known as an *impulse turbine.* The other kind is known as the *reaction turbine*, inasmuch as its rotor is caused to move in the direction opposite to that which a jet of steam issues tangentially from its circumference. Such a machine moves in consequence of the reaction of the flowing steam, and was the well-known steam engine of Hero of Alexandra, 130 B.C.; it is analogous to Barker's mill in hydraulics.

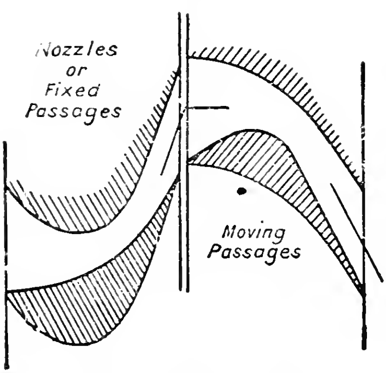


Fig. 56.—Reaction Turbine.

The Modern Reaction Turbine has, as a rule, no actual nozzles, but their equivalent is provided by making the blades of a special section and setting them at such an inclination, as shown in fig. 56, from these channels the steam issues at so fine an angle of inclination that the component of the reaction resolved tangentially is large, and causes it to turn. The guide channels or the equivalent of nozzles, through which the steam is led into the rotor, may be formed in the same way as shown in this figure.

The Modern Impulse Turbine also may have, and generally has, a series of expanding passages, the equivalent of nozzles formed by the blades in a similar way, or they are formed in a casting and fixed to the circumferential part of the stator, or fixed disc, attached to the turbine casing, as in fig. 57, which is an example of passages made in that way.

Combination Turbines.—For many reasons the efficiency of the pure impulse and pure reaction turbine is too low for commercial success, so that all modern turbines are so constructed that both the elementary processes of impulse and reaction occur within each pair of stator and rotor sets of blades. Their construction has developed along two characteristic lines, in which the impulse or reaction process predominates respectively. In what is now called the *impulse turbine* expansion of steam and generation of velocity take place solely in the fixed nozzles; the issuing jet acts directly on the leading half of the rotor blades, giving up part of its energy to them, and being then deflected through an angle of nearly 180° gives up the remainder by reaction. There is no change of pressure within the cell containing the moving blades, nor in the moving passages, except what is due to friction.

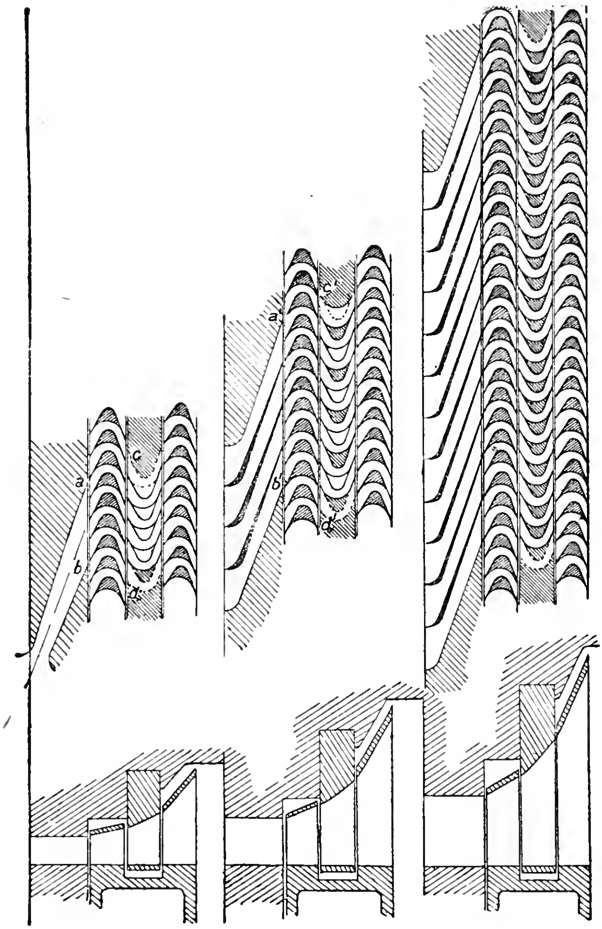


Fig. 57.—Impulse Turbine Compounded for Pressure and Velocity.

Consequently the nozzles may be isolated or in groups (*v.* fig. 59). *The reaction turbine*, as now made, is not unlike the above, but in it there is provided a further drop in pressure, and the corresponding generation of velocity in the moving passages. In this case, since there is a drop in pressure in these moving passages, the fixed passages or nozzles must occupy the whole of the annulus, in order to avoid excessive leakage by short-circuiting.

The Shape of Passages, and consequently that of blade section, depends on the drop of pressure required. If that drop is greater than p_1 to $p_2 = 0{\cdot}58\ p_1$, the passage should converge and then diverge; if p_2 is greater than $0{\cdot}58\ p_1$, convergent only. Thus in the example given (fig. 56) of simple turbines, large drop is provided for, and the passages are, therefore, convergo-divergent. Marine turbines being compounded by numerous stages, and having to run at low velocity, require convergent passages. The reaction turbine, having for convenience a similar ratio of expansion for each fixed and moving row of a given diameter, will have similar blading for both.

Compound Turbines of both types may be multiplied in series, limiting the drop in pressure for each stage to a portion of the total "head." This is called *compounding for pressure.* This compounding of the impulse turbine may be supplemented by another kind, in which the velocity of the nozzle jets is expanded in more than one row of moving passages, the velocity of

Fig. 57*a*.

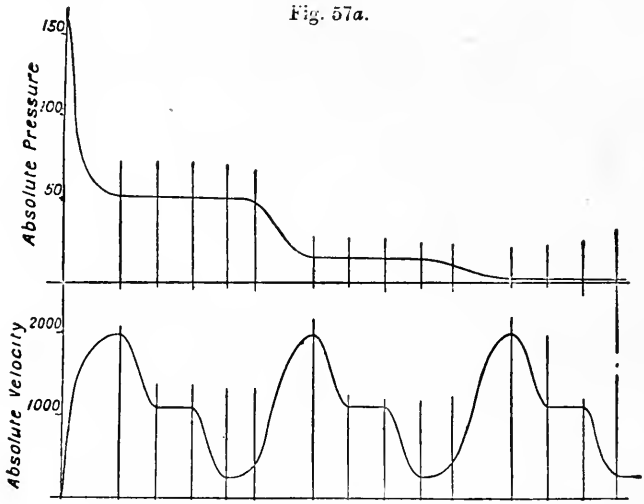


Fig. 57*b*.

the latter being less than in the uncompounded stage. This is called *compounding for velocity.* Fig. 57 shows diagrammatically an impulse turbine compounded for pressure in three stages, each being compounded for velocity once. The velocity changes of the steam are shown in fig. 57*b*, and it may be noted that the only function of the row of fixed blades between the moving rows is to deflect the stream.

An impulse turbine compounded for pressure may have a series of rotor discs with blades set transversely and interposed between a series of stator discs having guide blades forming a set of nozzle-like channels. If these blades are all of the same height, gradual decrease in pressure through the series is provided for by starting with only a few passages in the first stator disc, the number in the next disc is larger, and a further increase made in the third, and so on, the increase being such at each step that the steam expands in accordance with the natural law.

Expansion, however, may be provided for in quite another way; the length of stator and rotor blades may be gradually increased in length, so that variation in area of the anulus is such as to permit of the steam expanding in the same way while it follows its course through the machine. In either case there is at each drop in pressure a corresponding generation of velocity at each stator, and a delivery of kinetic energy to each rotor disc, so that the expansion is continuous step by step from the initial pressure at entry to that at exhaust, and the total work done distributed through the whole of the stator discs.

The object of this system of compounding is to reduce the mean velocity of flow to a minimum, so that the rate of rotation of the machine may be reduced down to practicable limits. So far as marine work is concerned. those limits must be such that a reasonable efficiency may be realised with the propellers revolved direct by the turbine.

The impulse turbine is not necessarily compounded for pressure; it may be compounded for velocity a sufficient number of times to use up that generated in *one drop* from maximum to minimum pressures. The number of velocity stages, as they are called, depends on the velocity of the blades relative to the initial velocity of the jets. On account of the losses by friction and eddies this type is not commercially efficient; if this were not so they would displace all other types for large sizes, inasmuch as they would be of less size, lower cost, and freer from the mechanical troubles so common in this instrument.

The turbine, however, may be treated in just the same way as a compound or triple-compound engine by providing means whereby there are two or three separate stages in the operations of the machine, in each of which there is at commencement a partial drop of pressure with a corresponding generation of velocity, followed by the gradual drop in velocity and the imparting of kinetic energy by the usual steps above described; at the end of the first stage, and before beginning of the second, there is another drop in pressure due to the passage through the second set of guide and expansive nozzles of larger capacity than the first; the steam acquires thereby a fresh velocity, and gives up its energy step by step through the second stage; and so on in a similar way through a third of similar construction, until the steam exhausted of practically all its potential energy enters the condenser.

Fig. 57*a* shows diagrammatically the fall in steam pressure in a three-stage compound-impulse turbine, and fig. 57*b* the corresponding generation of velocity and its gradual extinction in the two steps in each stage, as shown in fig. 57. In practice there are usually more than two steps in each stage of a turbine, and on shipboard it happens usually that there is a high-pressure turbine driving its own propeller, and exhausting to one and generally two low-pressure turbines, each with its own line of shafting and propeller. There may be, however, three turbines in series, high, medium, and low pressures, as in a triple-reciprocating engine, each with its own line of shafting and screw. These arrangements are necessary for marine purposes that the revolutions may be as low as possible to admit of screws of such a reasonable diameter as to be serviceable and at the same time efficient. Fig. 58 shows a section of the Ljungström turbine with its two interlaced rotors, which move in opposite directions and irreversible Fig. 59 shows in section a Curtis L.P.

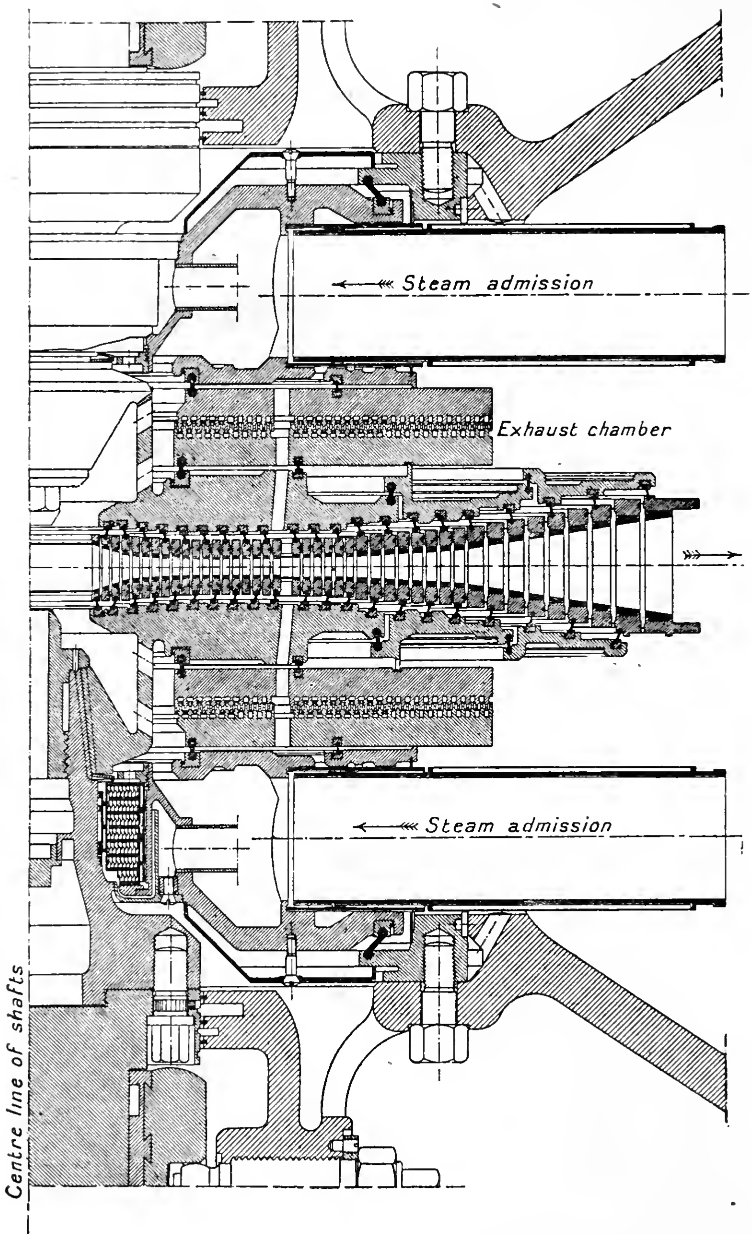


Fig. 58.—Ljungström Turbine (Longitudinal Half-section).

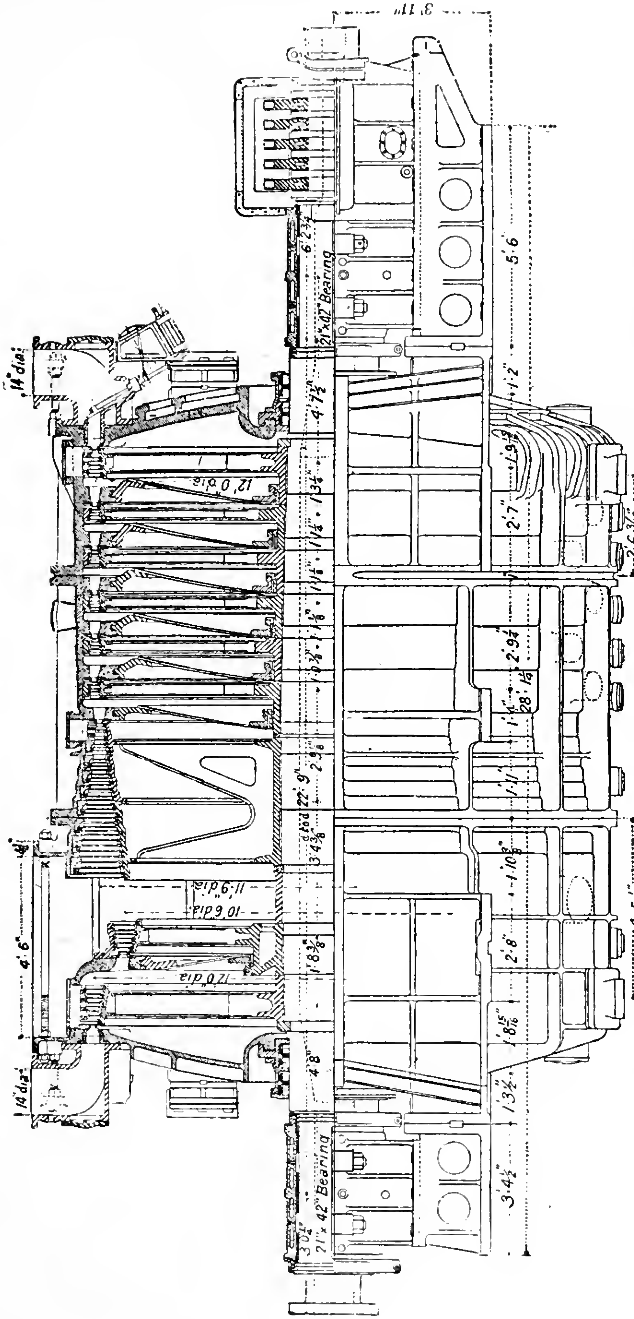


Fig. 59.—15-Stage Curtis Turbine of a Twin-Screw Japanese Battleship, 27,000 Collective S.H.P.

and astern-going turbine for a warship. Figs. 59*a* and 59*b* give the section and elevation of the Zoelly turbine, as used in the German Service.

The Design of Screw for a Turbine Ship must first of all be determined from the conditions imposed by the ship and her service, and the turbines designed to suit the revolutions at which such screws can be run consistent with good efficiency and safety.

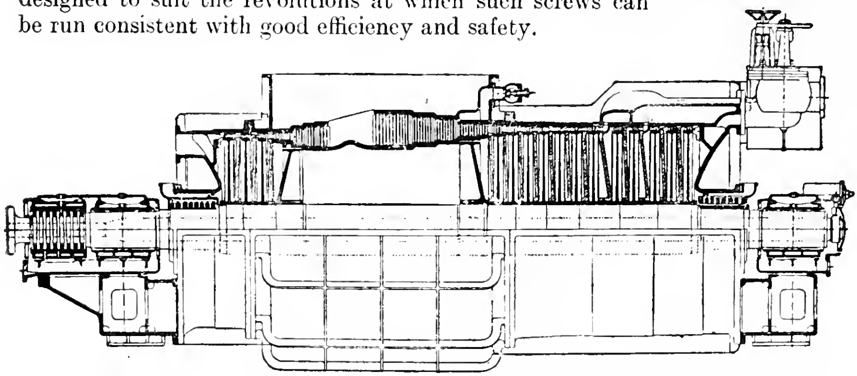


Fig. 59*a*.—Zoelly Marine Turbine. 7,000 H.P. at 650 r.p.m.

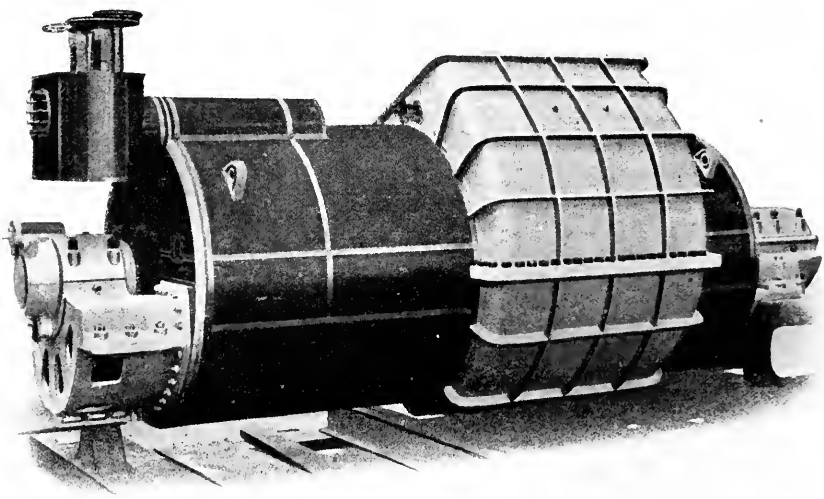


Fig. 59*b*.—Marine Turbine of 7,000 H.P. (Zoelly.)

The Efficiency of the Turbine* must, therefore, be determined at varying revolutions near to that rate at which the propellers may be run, and a curve plotted in the way usual to express it. The efficiency of the screw decided on by considering the above-named conditions should be plotted in a similar way and the curve formed, after which the combined efficiency can be calculated and expressed by its curve, and from it the designer can definitely and finally decide the exact speed of revolution at which the machines shall run.

* It is only with direct-driven turbines that these considerations require to be made (*v.* Appendix A).

Power developed by a Turbine cannot be obtained direct from it, as that of a reciprocator is by means of the indicator, so some other means of finding it has to be adopted. When driving a dynamo to generate electrical current it is easy to determine the power exerted by a turbine by means of the electrical instruments used for measuring quantity and intensity of current. From data obtained in this way turbine efficiency has been deduced so that if the weight of steam used can be accurately determined, the power output can be calculated. To-day, however, marine engineers have a better means of gauging the capacity of the instrument driving the propeller in the torsion meter than even the indicator ever was.

Prof. Rateau's Formula for Steam Consumption is as follows:—

$$\text{Steam consumption in a turbine} = 2{\cdot}13 + \frac{16{\cdot}2 - 2{\cdot}05 \log P}{\log P - \log p}.$$

P is the initial pressure absolute. p is the terminal pressure absolute.

The Sea Experiences with Turbines in the cruiser "Chester," as related by Lieut. Yates, U.S. Navy, are interesting as well as instructive. This ship is fitted with Parsons' turbines of 19,000 S.H.P., driving four screws; she attained a speed of 26·52 knots on trial. The lieutenant begins by emphasising the extreme importance of warming up the turbines thoroughly before attempting to run them, otherwise there is great danger of stripping the blades. This operation takes much longer than with a reciprocator, and should be from 3 to 3½ hours when possible; the turbine can be, of course, warmed quicker, but it is not advisable to do so.

Water in the turbine causes no apprehension, and quickly disappears. "Whipping" of the rotors occurs in the H.P. turbine in bad weather, causing it to vibrate and groan slightly, and when the astern-going turbine is operating there are tremors.

Manœuvring was not difficult, for as many as 85 signals from the bridge were responded to in 50 minutes. It is found that the turbines do not adapt themselves to the new conditions imposed on them consequent on changes of speed made during manœuvres at sea.

A drop in boiler pressure from any cause generally is followed by wet steam, and the increase in blade friction due to it; the loss of speed is thereby aggravated; and a drop of 1 inch of vacuum caused a loss of speed of half a knot. Great care is necessary with the bearings, for if they get warm water cannot be applied as with the reciprocator; there is nothing for it but to slow down, and sometimes to stop altogether, for a little heating causes sufficient expansion to make things much worse very quickly. The oil served, therefore, should be most carefully examined and kept in a high state of efficiency, and the oil used should be of the best and fittest for such service. Hot thrust bearings were not uncommon, due to the fine adjustments necessary to them. The wearing down of the bearings also is the cause of much trouble, even though it be as little as $\frac{5}{1000}$ of an inch, for then the gland packings leak and make the engine-rooms unbearable if the ventilation is not really good. If the propeller blades are injured, and even slightly deformed, there is a marked effect on the performance; but no cavitation was observable in this ship. Seeing, however, that the screws were quite small in diameter and well immersed, it is not very astonishing.

The mechanical efficiency of these turbines was very satisfactory, for

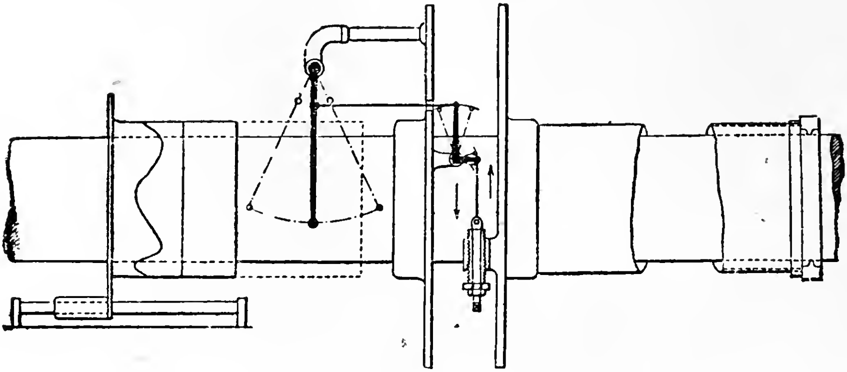


Fig. 60.—Fottinger Torsion Meter

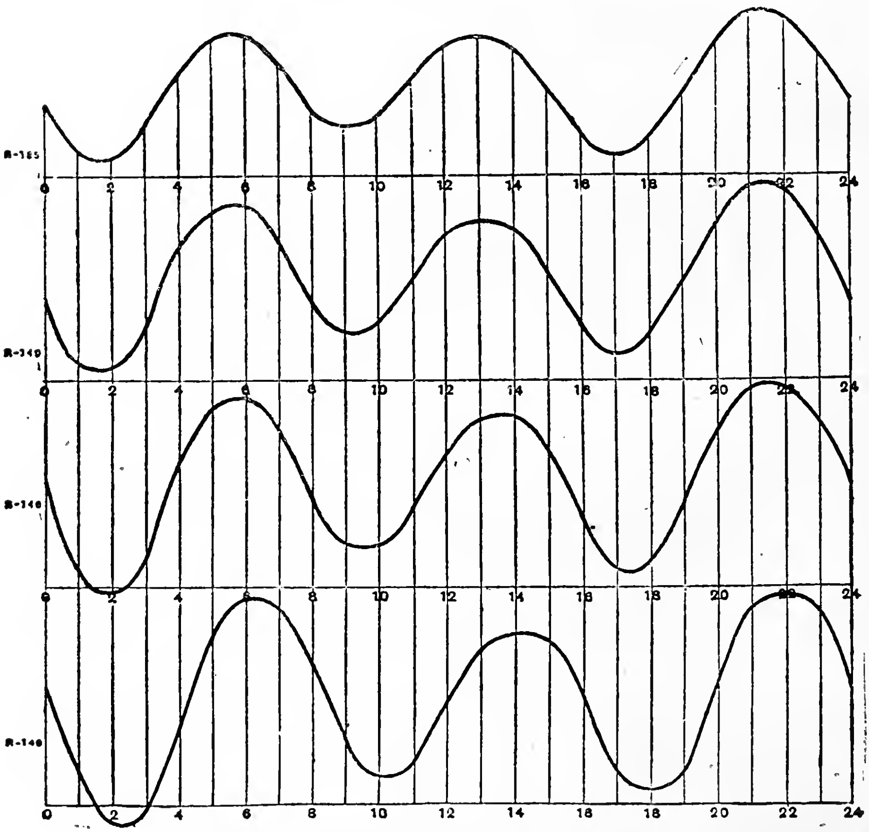


Fig. 60*a*.—Fottinger Torsion Meter Diagrams (Reciprocator).

when there was a small leak at the main stop valve of one, it kept turning at 100 revolutions per minute, while the vacuum held good; and with steam shut off and no vacuum another shaft ran at half the revolutions of those then engaged in propelling the ship, due to "drag" on its propeller.

Torsion Meters are now used on all trials of marine turbines, and by the engineers in charge of the machinery. As the name signifies, they are the measurers of the torsion or twist of a shaft—that is to say, they register the relative angular movement of two transverse circular planes at a definite distance apart by a line in a plane through the axis, thus noting the points on their circumference and their distance apart when the shaft is subject to torsion. Suppose two circular discs to be keyed on a shaft 10 feet apart; they are of considerable diameter, so that any small angular movement gives an appreciate circumferential one. If each has a radial line marked on its face so that these lines will be the traces of a plane through the shaft's axis when transmitting no power, then when transmitting power they will be no longer in line one with the other, for the axial plane through the one line will be at an angle with that through the other. Moreover, experiment has shown that the angular movement will be in proportion to the amount of torque and, therefore, is a measure of it. Now, suppose one of the discs to have a light metal cylinder fitted to it, of sufficient length to come close to the other disc without actually touching it, and a line is marked on this cylinder longitudinally to indicate the position of the disc's radial line, any torsion of shaft will be seen at once by its displacement past the marking on the other disc. In a general way this displacement could not be observed when the shaft is revolving, and certainly could not possibly be noted at the speed of revolutions of turbines.

Amlser's Torsion Meter, however, is constructed on this principle, and its indications are easily and clearly seen and read by the ingenious method of causing an electric spark to give an instantaneous momentary illumination of the index as it passes the eye, which produces the effect of an apparent stoppage of revolution, and admits of the reading of it quite easily.

Dr. Fottinger's Torsion Meter is a mechanical apparatus, so arranged that it can register its movements on a sheet of paper, as does the common indicator. In this case (fig. 60) there is a mechanical connection with the two discs and sleeve by means of a system of compound levers, the outer end of the last one having a pencil or tracing pin, which rests on a sheet of paper laid on a fixed cylinder surrounding the shaft. If no power is being transmitted a plain line is made on the paper as the shaft revolves; when power is being transmitted by a reciprocating engine a wavy line some distance from that base line is traced (*v.* fig. 60*a*), the waviness being due to the variation in twisting moment during each revolution of such an engine. With a turbine under load there is no such variation in smooth water, consequently the meter traces another plain line parallel to the first, and its distance from it is the measure of the torque. The accuracy of this instrument depends very much on that of the mechanism, and seeing that what it rests on is in motion all the time the wear on it due to this will not improve it in that respect. Nevertheless, effecting, as it does, its own register, makes it a very useful instrument for observation, more especially is it the case when applied to reciprocators. There are, however, other instruments in general use which have very little or no mechanism, and such as there is permits of no doubt to arise as to the accuracy of the results derived from its use.

The Hopkinson-Thring Torsion Meter has the great advantages over any apparatus produced hitherto that it requires only a very short length of shaft, and gives a direct reading of the torsion on the shaft. The zero point of the apparatus also can readily be ascertained by "barring" the shaft. The instrument offers also a simple means of determining the friction of the shafts themselves.

The principle of the apparatus designed by Professor Hopkinson and Mr. Thring is a differential one, and consists in the observation of the twist between two adjacent points on the shaft by means of two beams of light projected on to a scale from a fixed and a movable mirror. The beam projected on the scale by the fixed mirror is taken as the zero point, whilst the beam projected by the movable mirror indicates the amount of torque on the shaft. Both mirrors revolve with the shaft, but even at moderate

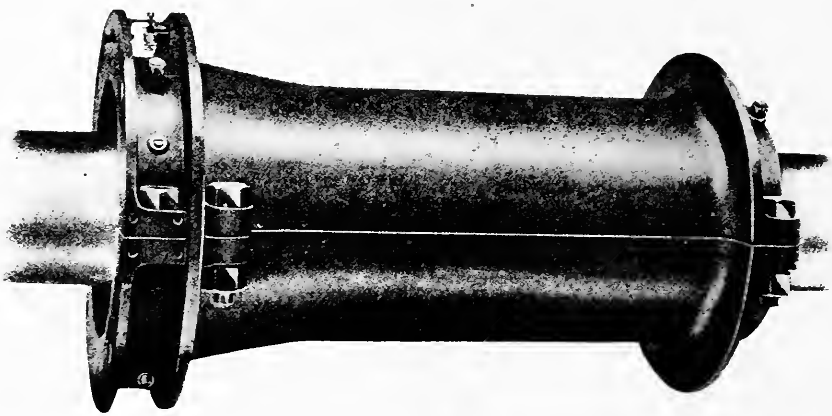


Fig. 61.—Hopkinson and Thring's Torsion Meter Mounted Complete on a Shaft.

speeds the reflections appear as continuous lines of light across the scale, and there is, therefore, no difficulty in taking readings.

The torsion meter is shown in fig. 61, mounted complete on a shaft, whilst a diagrammatic arrangement of the complete apparatus is shown in end elevation and plan in fig. 61*a*. A collar, A, clamped to the shaft of which the torque has to be measured, is provided with a flange projecting at right angles to the shaft and an extension.

A sleeve B (fig. 61*a*), provided with a similar flange and extension at one end, is clamped at its further end on to the shaft in such a manner that its flange is close to that on the collar A, whilst its extension overlaps that of the collar A, on which it is supported to keep it concentric. Both the collar and sleeve are quite rigid, and it is, therefore, obvious that when the shaft is twisted by the transmission of power, the flange on the sleeve B will move relatively to that on the collar A, the movement being equal to that between the two parts of the shaft on which these fittings are clamped. This movement is made visible by one or more systems of torque mirrors mounted between the two flanges, which reflect a beam of light, projected from a lantern, on to a scale divided in a suitable manner on ground glass.

Each system of torque mirrors consists of a mounting, pivoted top and bottom on one or other of the flanges, in which two mirrors are arranged back to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn the torque mirror, and thereby

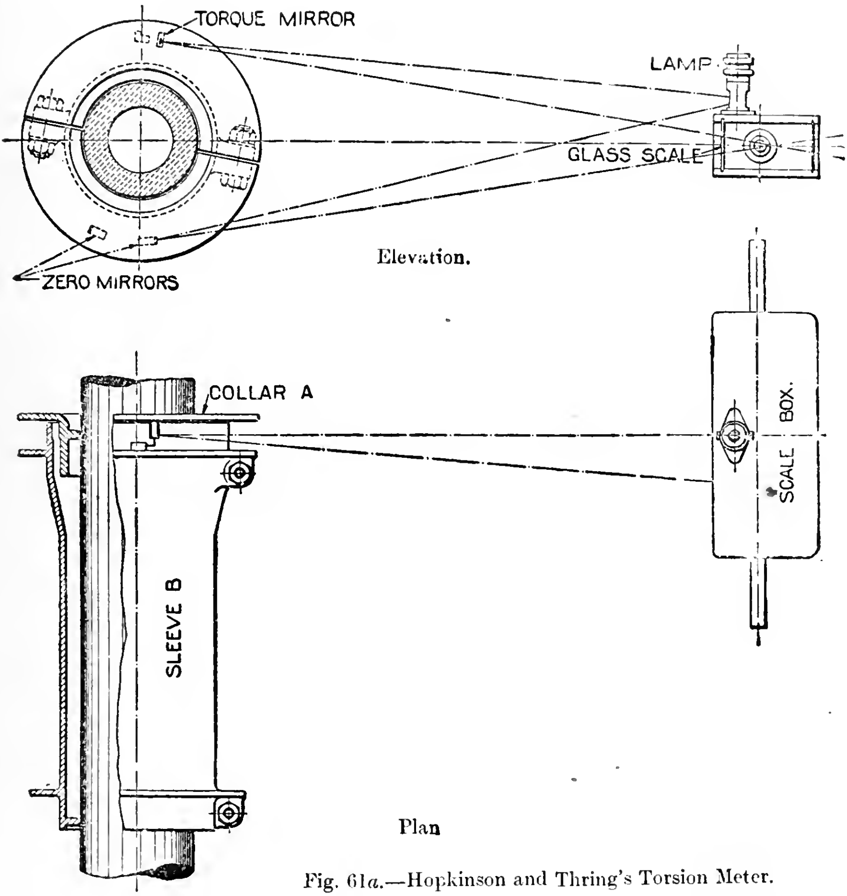


Fig. 61*a*.—Hopkinson and Thring's Torsion Meter.

cause the beam of light to move on the scale, the deflection produced being directly proportional to the torque applied to the shaft.

With the arrangement described, a reflection will be received from each mirror at every half revolution of the shaft; but where the torque varies during a revolution (as with reciprocating engines), a second system of mirrors may be arranged at right angles to the first system, so that four readings can be taken during one revolution; or, if two scales are used, eight readings.

Fig. 61*a* shows how the beam of light reflected by the mirror when in its highest position passes through the upper part of the scale; while the second reflection will occur when the mirror is in the position occupied by the zero mirror, the beam of light passing through the lower part of the scale. The position of the torque mirror in plan is such that the reflected beam strikes the scale to the right of the zero line, but when the shaft has made a further half revolution, the reflected beam from the other mirror will strike the scale to the left of the zero line. Obviously the deflection on both sides should be equal.

The fixed mirror is attached to one of the flanges (in fig. 61*a* to the flange of the sleeve B). This must be adjusted so that the beam of light reflected from it is received at the same point on the scale as those from the movable mirrors when there is no torque on the shaft. To facilitate the erection and adjustment of the apparatus, the box containing the scale and carrying the lamp is fitted with trunnions, so that it can be inclined as required.

If the position of the apparatus becomes altered relatively to the scale owing to the warming up of the shaft or from other causes, this is indicated immediately to the observer by an alteration in the position of the zero as reflected by the fixed mirror. Hence, the zero can be adjusted by moving the scale so that its zero coincides with the reflection from the fixed mirror. It will be obvious that it is not necessary to move the scale, as the mean of the two readings will be the same. It will readily be understood that a movement of the torque mirrors can only occur through a relative movement of the two flanges, so that vibration of the shaft or of the ship will not influence the readings in any way.

The constant of the instrument—viz., the factor which, when multiplied or divided into the product of the torsion-meter reading and the revolutions—gives the horse-power, may be calculated within 2 or 3 per cent., if the section of shaft within the instrument is uniform. But a direct calibration of the shaft with the instrument in position before the former is put into the ship should be made. This is effected easily by applying a known twisting couple. It is no inconsiderable advantage of this instrument that a direct calibration is established between the torsion-meter deflection and the torque on the shaft.

In the Bevis-Gibson Torsion Meter, which is largely used in every-day work, light is also used to indicate the angular movement. In this case however, the discs are quite a part, and perforated with small narrow slits in such a way that when the shaft is running without load the light of an electric lamp behind the one disc can be seen through a sighting instrument, called a "finder," behind the other disc—that is, the lamp slit, its disc slit, the other disc slit, and the "finder" slit are all in line every time that part of the discs pass, when the shaft is running without load, so that on looking through the finder the flash of light is seen every time the slit passes the finder; and since the revolution is rapid, it appears as a continuous illumination. When power is being transmitted, the shaft is twisted; the slits get out of line, and no light is seen through the "finder." The latter, however, can be moved through an angle by means of a finely adjusted screw, etc., until the light is again picked up when looked at through the "finder" disc's slot at the time the lamp's disc's slot is passing the light slit. The angular displacement of the "finder" corresponds with the angular movement of the shaft, and the shaft horse-power can be determined thereby in the usual way.

Collie's Torsion Meter (fig. 62) is also a mechanical contrivance of considerable ingenuity, by which the angle of twist is caused to be registered by a pointer and dial, not unlike the Bourdon pressure and vacuum gauge used on the L.P. valve box of a compound engine. In this arrangement two counter shafts are connected at the middle by the coarse-threaded end of one entering the threaded sleeve turned by and free to slide on the end of the other. Each shaft is driven independently by a Renald chain geared up so that it runs about three times to one of the main shaft. If the main shaft is not transmitting power, the sleeve simply revolves, and the pointer

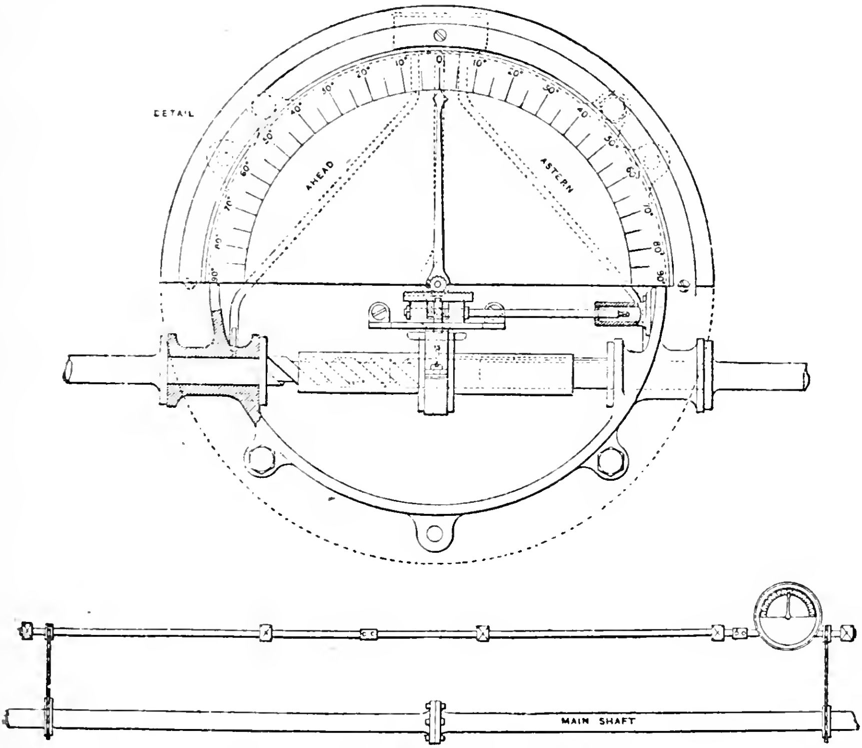


Fig. 62.—Collie's Mechanical Torsion Meter.

of the gauge is motionless in mean position. As soon as power is transmitted the shaft twists and one counter shaft moves in advance of the other, and forcing the sleeve to slide as a consequence of its screwed end moving on the male end of the other; the pointer is thus caused to turn round so as to indicate the exact angular displacement of the length of shaft between the two driving wheels. Both these two meters, as, indeed, must all mechanical ones, depend on the accuracy of manufacture and the state of repair for the value of their indications; it is evident that the magnification of the errors will be practically at the same rate as the magnification of the indications. It was natural, therefore, for scientists and inventors to turn to the other means of indications and magnification, which had provided the means for measuring other forces with a delicacy that no mere mechanical contrivance can achieve. The mirror magnification of radius and consequently of the arc used in telegraphy no doubt suggested itself to others as it did to Prof. Hopkinson and Mr. Thring.

The Denny-Johnson Meter differs from the others, inasmuch as linability of the points on the discs or otherwise is indicated by sound, instead of light, as transmitted to a telephone receiver. When the two points or projections in this case on the disc are in line, they are so close to two fixed projections as to make virtually an electrical connection. When torsion takes place the connection is broken—that is, the meeting of the projections do not synchronise—and it is only by displacement of one of these fixed points equal to the relative circumferential movement of the point on the disc that restores synchronising contact with the consequent production of sound in the telephone. The amount of this movement is registered in the ordinary simple way, and from it the horse-power is estimated.

Shaft Horse-power transmitted by a shaft can be calculated from the *torque* as follows :—

T is the twisting moment or *torque* in inch-pounds.

R, the revolutions made per minute by the shaft.

$$\left.\begin{array}{r}\text{The work performed}\\ \text{per revolution}\end{array}\right\} = \frac{2\pi \times T}{12} \text{ or } 0{\cdot}5236\text{ T.} \quad \text{S.H.P.} = \frac{0{\cdot}5236\,T \times R}{33{,}000} = \frac{T \times R}{63{,}000}$$

The torque can be calculated from the angle of twist or torsion by means of the following formulæ :—

a is the arc at a radius r of the angle of torsion θ, $\frac{a}{r} = \beta$; l is the length and d the outer and d_1 the inner diameter of shaft.

$$\beta = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4} \text{ (Rankine).} \qquad \frac{\theta}{360} = \frac{a}{2\pi r} \text{ or } = \frac{\beta}{2\pi}.$$

$$\text{That is } \beta = \frac{\theta \times 2\pi}{360} = \frac{\theta}{57{\cdot}3}. \qquad \text{Then } \frac{\theta}{57{\cdot}3} = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4},$$

$$\text{or (i.) } \theta = \frac{584 \times T \times l}{M_r \times d^4}\left.\vphantom{\frac{1}{1}}\right\}\text{for solid shafts,} \qquad \text{(ii.) } \theta = \frac{584\,T \times l}{M_r(d^4 - d_1^4)}\left.\vphantom{\frac{1}{1}}\right\}\text{for hollow shafts.}$$

$$\text{That is} \qquad T = \frac{\theta \times d^4 \times M_r}{584 \times l}.$$

M_r is the modulus of stiffness or rigidity of the material, which for steel generally is 10 to 12 millions. With steel shafts of best make, experiments have shown the value of M_r to be 11,750,000 for solid, and 12,150,000 for hollow. In every-day practice, 11,250,000 is taken for solid steel shafts :—

$$\text{Then} \qquad T = \frac{\theta \times d^4 \times 11{,}250{,}000}{584\,l} = 19{,}264\,\frac{\theta \times d^4}{l}.$$

Substituting this value of T in the formula for S.H.P.

$$\text{S.H.P.} = 19{,}264\,\frac{\theta \times d^4}{l} \times \frac{R}{63{,}000} = \frac{\theta \times d^4 \times R}{3{\cdot}27 \times l} \text{ solid shafts.}$$

$$\text{S.H.P.} = \frac{\theta\,(d^4 - d_1^4)\,R}{3{\cdot}17 \times l} \text{ for hollow shafts.}$$

It may be observed that it is usual to experiment in the workshops with the shafts of every ship to ascertain beforehand what amount of torque is necessary to produce a degree of angular movement in a definite portion of its length; from such experiments the modulus of rigidity is ascertained. It is then easy to construct a diagram from that particular shaft, by which the power may be read off for any angular movement indicated by the torsion meter, and the revolutions at the time of observation. Fig. 63 is such a diagram as to need no explanation.

It was held at one time that the end pressure on a ship's shaft due to thrust seriously affected the register of torque; it was supposed until lately to affect it to the extent even of 3 per cent.; but more recent investigations by Dr. Hopkinson with more sensitive instruments seem to have removed the impression, and that practically end thrust may be disregarded now as a factor in the calculation of shaft horse-power.

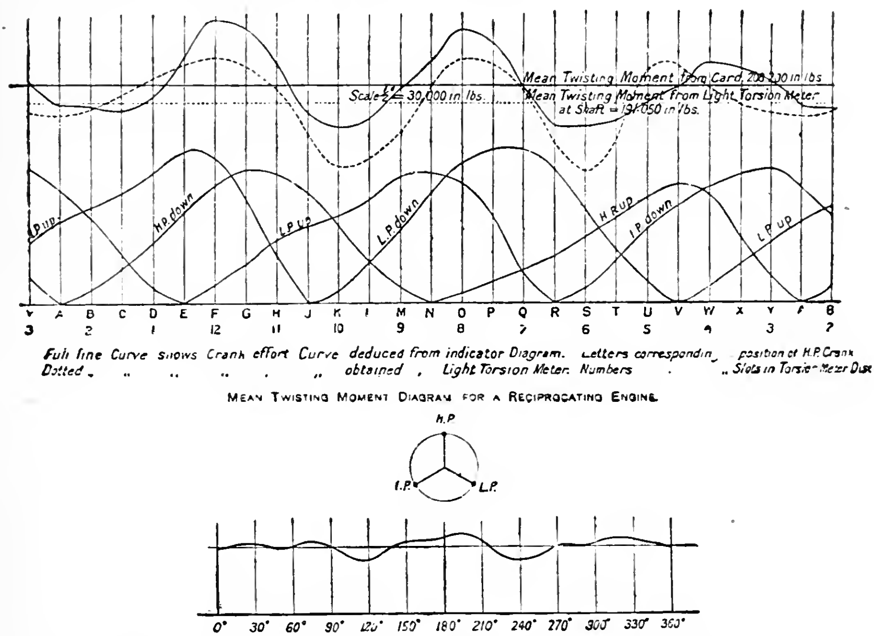


Fig. 63.—Crank Effort and Torsion Meter Diagram (J. Hamilton Gibson).

The Ordinary Steam Engine Indicator is often spoken of as a defective instrument, and one not to be relied on to give accurate results. It is quite true that it is not exactly a perfect one, and that the power registered by means of it comes short of the actual amount developed in the cylinders, but, at the same time, while admitting this, it should not be forgotten that it renders another and a very good service to the engineer besides that of giving him the power, and one that is quite as important to him. It shows in a ready and rapid way whether the internal and unseen parts of the engine are in good working order, and efficient for their service. It is to him what the stethoscope is to a doctor of medicine.

CHAPTER VI.

EFFICIENCY OF MARINE ENGINES.

The Efficiency of an Engine should be measured by the cost of the useful work it does on service; formerly the fuel burnt in the boiler from which the steam was supplied was taken as the measure of cost, and so long as the fuel was of standard quality and the efficiency of the boilers constant, it was a fair and ready means of doing so. But coal from the same pit may vary in calorific value even when freshly raised, and undoubtedly does so after exposure; the method of ascertaining the weight permits of inaccuracy, although perhaps this is only slight; the state of the atmosphere seriously affects consumption, for it is obvious that cold moisture laden air will require more fuel to raise it to the temperature of the furnace for combustion than is necessary for an equal amount of dry warm air; the human element likewise plays an important part in the production of steam, as may be evidenced when the single boiler of a ship is barely large enough for the engine requirements; in such a case a good and experienced stoker will keep the steam pressure and supply steady, whilst an indifferent one will use more fuel, and then fail to maintain full pressure. Further, atmospheric conditions seriously affect the quality of the draught, so that while with a good breeze the draught will be sharp and the combustion efficiently effected, on a hot sultry day with no wind it will be bad, and the fires require much forcing. Then, too, the condition of the boiler is not always the same; the grate bars may be faulty, the tubes dirty, and the inside with more scale at one time than another; but it is only fair to say that nowadays the scale should never be so very great as to cause serious differences. Taking all these things into account, it is obvious that the more correct method is to debit the engine with the weight of steam supplied to it rather than with the weight of coal. In old days, when there were no auxiliaries, but only the main engines, there was no call to differentiate as there is now the steam supplied from the boilers between that used by the main engine and that by auxiliaries, etc. To-day the amount needed for other purposes than that of driving the main engines is very appreciable, amounting to as much as 14·5 per cent. of the total used in the "Lusitania" when running at full speed. There are in passenger ships centrifugal circulating and air and feed pumps constantly going, but they are virtually a part of the main engines; steering engines, ash hoists, blowing fans, ventilating fans, refrigerators, electric lighting, sanitary pumps, etc., are also working during a considerable portion of the day, and bilge pumps, fire and wash-deck pumps, steam whistle, etc., take a good share also; besides all these in cold weather steam heating will demand an extra expenditure of fuel, an important item in the North Atlantic.

The main propelling machinery, therefore, must be debited only with the steam it uses and that used by such of the auxiliary machinery as is necessary to keep the main engines running.

With such feed and other pumps as are now employed in the engine-rooms of important vessels, the water consumption can be closely measured, as they are capable of acting as meters of the water they pass by simply fitting them with counters to record the number of strokes they make; the chief engineer can thus quite easily calculate the actual amount of water pumped from the main condenser and from the auxiliary condenser, and record thereby the steam consumption of the machinery as a whole, or of the main engines only.

To show the cost of the *useful* work done by marine engines used to be somewhat difficult, and could be calculated only by using assumptions that were always somewhat uncertain. That is, until the torsion meter was used the work transmitted to the propeller was, except in some special exceptions, practically guessed at; now, however, thanks to these instruments, we can measure what is called the shaft horse-power of both turbines and reciprocators, and thereby compare the outputs by reducing them to a common denominator; this being so :—

Water consumed per S.H.P. (shaft horse-power) is the measure now of the efficiency of a marine engine, *quâ* engine, without complications or explanations. But by this method the general efficiency is shown without any differentiating between the steam and the mechanical efficiency of the system.

The Mechanical Efficiency, or the relative value of the engine as a piece of mechanism, is measured by comparing the shaft horse-power with the gross power generated, as shown by the indicator diagram;

Mechanical efficiency of a marine engine, therefore, is S.H.P. ÷ I.H.P.

Mr. Denny found by using a torsion meter that the mechanical efficiency of some quadruple engines made by his firm was as high as 94 per cent.

The late Mr. Mudd ascertained the power required to move certain triple-compound engines standing in the erecting shop without screw shafting was at working revolutions 45 I.H.P., or 5·0 per cent. only of the gross power indicated (900) when working at those same revolutions in a loaded ship. This would show, then, the efficiency to be 95 per cent.; but it must be noted that in this case no allowance is made for the extra friction on the valves, guides, etc., due to the greater pressure on them when at work at full power.

The older horizontal jet-condensing engines had comparatively a low efficiency due to the general friction of the very heavy moving parts and the resistance of the two air, two feed, and two bilge pumps worked by them. There is good reason to believe that the mechanical efficiency of some of them was seldom over 75 per cent. Latterly, however, the well-made horizontal engines of Penns, Maudslays, etc., when running at the higher revolutions, and developing much more power than formerly, and having surface condensers, had even 80 to 85 per cent. efficiency.

Vertical Engines with surface condensers and slow running had an efficiency from 5 to 8 per cent. better than the horizontals of similar size, and working under similar conditions. To-day naval machinery and that of express steamers have an efficiency from 90 to 94 per cent. at full speed.

when with only the air pumps driven by it, and of the triple- and quadruple-reciprocating type.

The Mechanical Efficiency of a Turbine has been determined approximately by observing the electro-motive power required to revolve it at working speed by means of a motor, and taking it as the mechanical loss when working. Its efficiency is, of course, high, and from such observations made with similar machines in service on land, it may be taken that the marine turbine has a mechanical efficiency of about 95 per cent. in a general way, and that with the largest ones it may have a somewhat higher, probably 96 to 97·5 per cent.

The Mechanical Efficiency of the Reciprocating Engine is not so high as this, although that of well-designed carefully made and balanced engines of high speed will not be far short of the 95 per cent. when running in good working order and free of all pumps, and well lubricated. Under these circumstances its efficiency is probably 92 to 94 per cent. at least. It is, however, somewhat misleading to express the mechanical losses as a fraction of the Total Indicated Horse-Power, inasmuch as that varies closely with the cube of the revolutions, while the losses vary more nearly as the revolutions; so that, although an engine may show an efficiency of only 80 per cent. at 50 revolutions, it may be as much as 94 at full speed of 75 revolutions —that is, the loss is $\left(\frac{50}{75}\right)^3 \times 20$—without any change in the adjustment of a single part.

Mechanical losses really depend largely on the size of the engine. Now, in a general way the Nominal Horse-Power expresses fairly well the size of any engine, and, therefore, all other things being equal, it will be a sufficiently accurate assumption that *frictional losses are proportional to N.H.P.* Practice has demonstrated that about 70 per cent. of these losses will in the ordinary marine engine vary directly with the revolutions within reasonable limits, and further that the remaining 30 per cent. increase at a more rapid rate; in fact, in modern engines total losses roughly vary as the cube root of the revolutions raised to the fourth power—that is, as $R^{\frac{4}{3}}$.

It is obvious that the mechanical efficiency of any engine will vary from time to time, and under some circumstance the variation may be very considerable. It is also well known that at very slow rate of revolution the apparent mechanical losses are uncertain, and always proportionately large. The only losses that can be considered here are those which inevitably occur with any engine when in quite a good state of repair and in really good working order.

The Efficiency of Marine Engines, when well made, in this good state of repair, and in good working order, should be approximately in accordance with the following rule :—

$$\text{Friction horse-power} = \frac{\text{N.H.P.} \times \text{R}}{1{,}000}(y + x\sqrt[3]{\text{R}}).$$

$$\text{Efficiency} = \frac{\text{I.H.P.} - \text{F.H.P}}{\text{I.H.P.}}.$$

$$\text{Nominal horse-power} = \text{D} \times \text{S} \div \text{K},$$

where D is the diameter of the L.P. piston, and S the stroke, both in inches.

For the two-stage compound engine K = 15·0.
,, triple-stage ,, K = 12·6.
,, quadruple-stage ,, K = 10·5.

R the revolutions per minute.

For diagonal paddle-wheel engines with air pumps only $x = 1·5$; $y = 10$.
,, vertical screw engines, mercantile, with all pumps connected, ... $x = 1·0$; $y = 8$.
,, light quick-running screw engines with all pumps connected, ... $x = 0·7$; $y = 7$.
,, naval and express screw engines, centrifugal circulating, ... $x = 0·6$; $y = 7$.
,, naval and express screw engines, with only air pumps $x = 0·5$; $y = 6·5$.
,, special naval screw engines, no pumps . . . $x = 0·3$; $y = 6·0$.
,, ,, forced lubrication, no pumps. . . $x = 0·1$; $y = 5·0$.

Example 1.—A "tramp" steamer having an engine with cylinders 22, 37, and 62 inches diameter and 45 inches stroke, which at 85 revolutions develops 2,300 I.H.P., what is the efficiency ?

Here $$\text{N.H.P.} = 62 \times 45 \div 12·6, \text{ or } 221.$$

$$\text{Frictional H.P.} = \frac{221 \times 85}{1,000}(8 + \sqrt[3]{85}) = 220.$$

$$\text{Efficiency} = (2,300 - 220) \div 2,300, \text{ or } 0·904.$$

Example 2.—A *destroyer* has two sets of engines, each having cylinders 19, 28·5, and 43 inches diameter with 18 inches stroke at 370 revolutions; the total I.H.P. is 4,200.

$$\text{Here N.H.P. each engine} = 43 \times 18 \div 12·6 = 61.$$

$$\text{Friction H.P.} = \frac{61 \times 370}{1,000}(6 + 0·3\sqrt[3]{370}) = 184.$$

$$\text{Efficiency} = (2,100 - 184) \div 2,100 = 0·913.$$

Example 3.—A paddle steamer having cylinders 56 and 110 inches diameter and a piston stroke of 72 inches develops 7,150 I.H.P. at 50 revolutions, what is the F.H.P. and efficiency ?

Here $$\text{N.H.P.} = 110 \times 72 \div 15, \text{ or } 530.$$

$$\text{F.H.P.} = \frac{530 \times 50}{1,000}(19 + 1·5\sqrt[3]{50}) = 411.$$

$$\text{Efficiency} = \frac{7,150 - 411}{7,150} = 0·942.$$

Example 4.—A naval ship has two engines, each with cylinders 43 and 69 inches, and two L.P. each 77 inches, the stroke being 42 inches.

Here the two L.P. cylinders are equivalent to one 109 inches diameter.

At 140 revolutions each engine develops 12,000 I.H.P.
,, 126 ,, ,, 8,000 ,,
,, 85 ,, ,, 2,500 ,,

Now, $\text{N.H.P.} = 109 \times 42 \div 12 \cdot 6 = 363.$

(a) $$\text{F.H.P.} = \frac{363 \times 140}{1,000}(7 \cdot 0 + 0 \cdot 6 \times \sqrt[3]{140}) = 514.$$

$$\text{Efficiency} = \frac{12,000 - 514}{12,000} = 0 \cdot 957.$$

(b) $$\text{F.H.P.} = \frac{363 \times 126}{1,000}(7 + 0 \cdot 6 \sqrt[3]{126}) = 460.$$

$$\text{Efficiency} = \frac{8,000 - 460}{8,000} = 0 \cdot 942.$$

(c) $$\text{F.H.P.} = \frac{363 \times 85}{1,000}(7 + 0 \cdot 6 \sqrt[3]{85}) = 298.$$

$$\text{Efficiency} = \frac{2,500 - 298}{2,500} = 0 \cdot 881.$$

Example 5.—A triple-compound engine with cylinders 18½, 28, and 40 inches diameter, the stroke 20 inches, and at 250 revolutions the I.H.P. is 1,150. There are no pumps, and lubrication is forced throughout.

Here $\text{N.H.P.} = 40 \times 20 \div 12 \cdot 6 = 63 \cdot 5.$

$$\text{Friction H.P.} = \frac{63 \cdot 5 \times 250}{1,000}(5 + 0 \cdot 1 \sqrt[3]{250}) = 89 \cdot 5.$$

$$\text{Efficiency} = \frac{1,150 - 89 \cdot 5}{1,150}, \text{ or } 0 \cdot 922.$$

Dr. Bauer states that from observations of and experiments with marine engines made in Germany the efficiency of small ones with an indicated horse-power not exceeding 50 is only 0·59, while those of 5,000 I.H.P. and upwards it was as high as 0·91; also that the efficiency varied roughly from 0·59 to 0·91 in accordance with their power for other engines. He quotes some examples of engines whose actual efficiency had been determined, and they are as follows:—

TABLE XXV.—EFFICIENCY OF SOME GERMAN ENGINES.

German Engine.	A	B	C	D	E	F
Indicated horse-power, .	1,630	1,640	1,940	2,370	2,890	4,500
Efficiency, ...	0·885	0·910	0·911	0·920	0·911	0·935

Mr. Denny found by torsion meter trials with a reciprocating engine that the efficiency at full power, 1,550 I.H.P., was 0·92; and in another case with a power of 1,950 it was as high as 0·935.

The Results of Trials made with Triple-compound Engines designed for electric generating are shown in fig. 64, and are very instructive, although not made under quite the same conditions as those obtaining with a marine engine, for in these cases the load was an artificial one (brake), and somewhat arbitrarily varied.

The following figures and diagrams show the results of some carefully made trials with two triple-compound engines, by Messrs. Belliss & Morcom, Birmingham. They are of their special enclosed type, having three cylinders and three cranks supplied with steam at 150 lbs. pressure, and exhausting into a surface condenser whose pumps are operated by independent engines. The whole of the pins, bearings, guides, and working parts are lubricated with their special American mineral oil forced through them by a pump constantly at work connected to the engine. It will be seen that the friction per revolution, both with and without load, varies, and has two minimum and two maximum values; that at full speed the frictional H.P. per revolution loaded is really only a little less than that when running free, whereas at 200 revolutions, or half speed, there is the greatest difference between them, that at full load being half that running free; also that at dead slow speed, say 25 revolutions, the friction per revolution, both light and loaded, is nearly four times that of the loaded engine at 200 revolutions.

ENGINE (No. 1401) $\dfrac{11'' - 17'' - 24''}{12''}$ 150 LBS. AT ENGINE STOP-VALVE.

Table of Powers, &c., Under Load Conditions.

	Revolutions.			
	166	300	350	400
I.H.P.,	248·2	384·15	408·1	429·4
B.H.P.,	240·4	360·5	380·5	400
Difference or friction, H.P.,	7·8	23·65	27·6	29·4
I.H.P. per revolution,	1·496	1·280	1·168	1·073
B.H.P. ,, ,,	1·45	1·201	1·086	1·00
F.H.P. ,, ,,	·047	·079	·079	·073
$\frac{\text{B.H.P.}}{\text{I.H.P.}}$ efficiency,	96·8 °/₀	93·9 °/₀	93·25 °/₀	93·15 °/₀

Friction Powers Under No-Load Conditions.

	Revolutions.					
	56	102	113	213	313	365
Friction H.P.,	4·9	10·7	7·4	24·6	36·1	30·9
,, ,, per revolution,	·0875	·1049	·0655	·1153	·1153	·0847

The above figures are embodied in the diagram (fig. 64).

The no-load cards were taken non-condensing, but the load cards were taken condensing.

This is probably the reason why the friction power shown at no-load

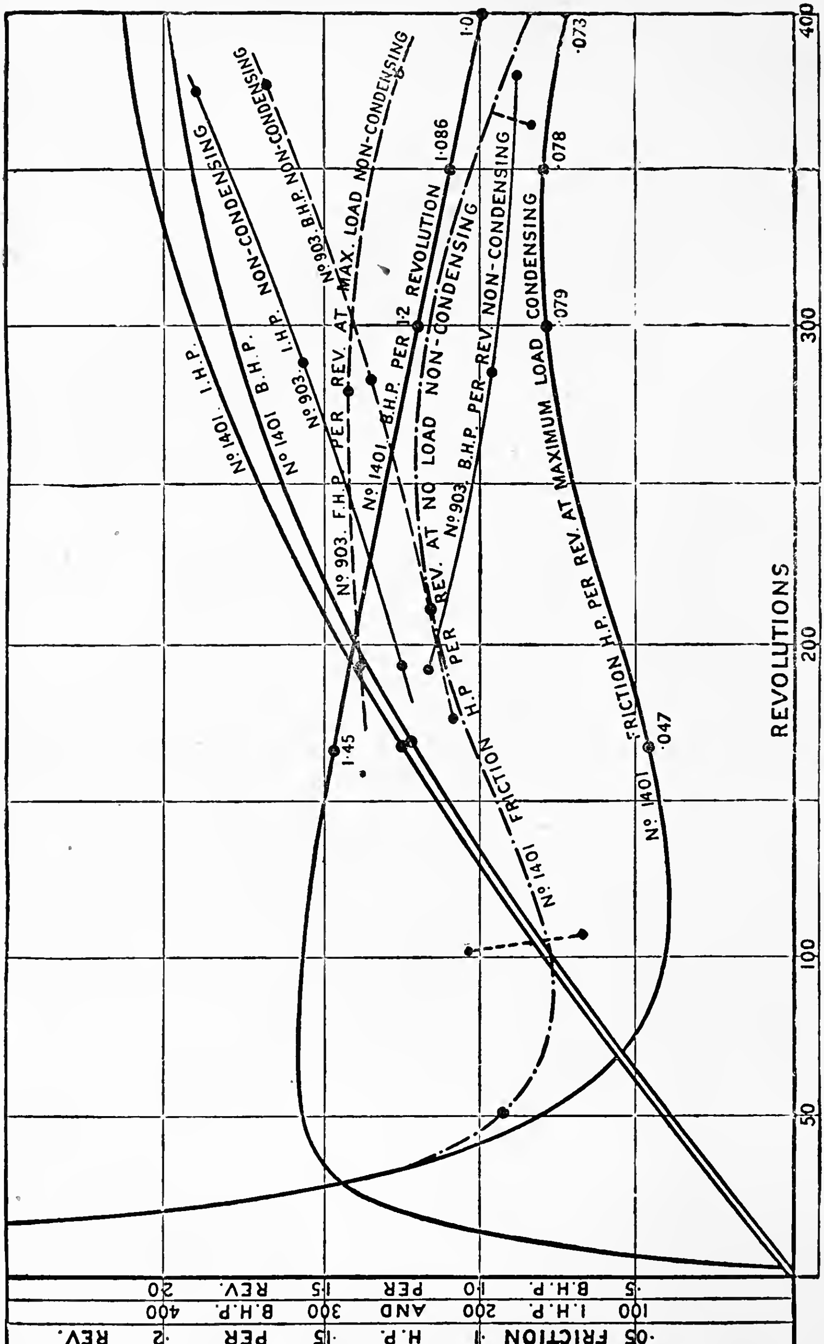


Fig. 64.—Friction Trials with a Triple-Compound Engine by Belliss & Morcom.

(1401) Cylinders 11 - 17 - 24 inches diameter × 12 inches stroke, steam 150 lbs., cut-off 0·8.
(903) ,, ,, ,, 10½ ,, ,, ,,

is greater than that at full-load. The efficiency under load condensing is invariably slightly better than when non-condensing.

It is also interesting to note how rapidly the rate of friction per revolution increased when the power was high and revolutions were quite small. The efficiency of the larger engine at full load and highest revolutions was 0·929, and the friction losses were ·047 I.H.P. per revolution at 170; 0·079 at 300 revolutions; 0·078 at 350 revolutions, dropping to 0·073 at 400. These, however, are engines having forced lubrication.

Mr. Yarrow's Experiments with a Torpedo Boat to ascertain the efficiency of engines, propeller, and ship were made in a most careful and exhaustive way, and are most interesting and instructive. They show that the mechanical efficiency of the engines varied from 0·766 at 9 knots to 0·923 at 15 knots, and that the mechanical losses varied at a higher rate than given by the arithmetical progression of revolutions. The high efficiency of these quick-running engines at full power when carefully manufactured and adjusted is manifested by an inspection of the figures in the subjoined table.

TABLE XXVI.—YARROW'S EXPERIMENTS ON EFFICIENCY.

Speed, - knots,	9·0	10·0	11·0	12·0	13·0	14·0	15·0
Ind. horse-power, .	38·5	49·5	67·1	99·0	143·0	193·6	255·2
Friction, etc., loss,	9·0	10·1	11·7	13·5	15·4	17·4	19·5
Efficiency, . .	0·766	0·796	0·827	0·864	0·892	0·910	0·923

Mr. A. H. Tyacke, of Hull, made a series of trials with the engines of two ships manufactured by Earles Co. to ascertain frictional resistance at different speeds. In each case the engines were allowed to run without their propeller shafting connected; indicator diagrams were carefully taken at various rates of revolution. At the same rates a set of diagrams were taken with the ship running with propellers connected. The first ship had cylinders 12, 20, and 32 inches diameter, and a piston stroke of 21 inches. The second ship had somewhat larger engines, the cylinders being $12\frac{3}{4}$, 22, and 36 inches diameter, and a piston stroke of 24 inches, and the highest revolutions only 111, which is small for such small cylinders. Under these circumstances the efficiency is remarkably good, especially that of the latter.

TABLE XXVII.—TYACKE'S EFFICIENCY EXPERIMENTS.

Revolutions per Minute. .	54	70	88	96	106
Indicated horse-power, . .	54·6	87·0	180·9	223·6	306·0
Frictional horse-power, .	20·5	27·4	34·7	40·4	46·2
Efficiency (I.H.P. − F.H.P.)÷ I.H.P., ...	0·634	0·689	0·808	0·819	0·849

TABLE XXVIIA.—TYACKE'S FURTHER EXPERIMENTS.

Revolutions per Minute,	40	60	93	111
Indicated horse-power,	38·80	82·20	322·00	480·60
Frictional horse-power,	14·15	22·39	40·70	47·05
Efficiency (I.H.P. − F.H.P.) ÷ I.H.P.,	0·635	0·728	0·873	0·902

Steam Efficiency is quite as necessary as mechanical efficiency to a successful marine engine, and has to be as carefully considered. First of all, the heat conditions under which it works are the raising of a pound of water from the temperature of the hot-well to a pound of steam with a sensible temperature, and a latent heat corresponding to the boiler pressure; during the cycle of operations on the engine it is reduced again to water of the original temperature. Its efficiency then will be measured by comparing the heat given up during the cycle when doing work with the total heat. If T is the temperature of steam at P pressure, L its latent heat, and t the temperature and l the latent heat due to the pressure in the condenser—that is, the total heat of evaporation at boiler pressure minus the heat of the feed-water is the amount debited, and that amount less the total heat of evaporation at the pressure in the condenser is to the credit account—the efficiency is, therefore,

$$\text{Efficiency of steam} = \frac{(\mathrm{T} + \mathrm{L} - t) - (l)}{\mathrm{T} + \mathrm{L} - t} = \frac{\mathrm{T} + \mathrm{L} - t - l}{\mathrm{T} + \mathrm{L} - t}.$$

The numerator of this fraction is the greatest amount of heat possible to be used with steam of the pressure, and multiplying it by 772 will give the maximum amount of mechanical work possible in an engine working under the conditions imposed. Dividing the product by 33,000 will give the horse-power.

For example, marine engine condensers may now be relied on to produce a vacuum of 28 inches, so that the back pressure is 1 lb. The total heat at this pressure is 102° sensible, together with 1,042° latent, or 1,144°. At 100 lbs. pressure absolute the total heat is 327° + 884°, or 1,211°; at 150 lbs. it is 358° + 862°, or 1,220°; at 200 lbs. it is 381° + 845°, or 1,226°, while at 250 lbs. it is 401° + 831°, or 1,232°.

The heat required to change a pound of water at 102° temperature to a pound of steam at these various pressures will be 1,109, 1,118, 1,124, and 1,130 B.T.Us.; in each case the rejection is 1,042°.

Hence at 100 lbs. pressure the heat available for power is 1,109° — 1,042°, or 67°.

The efficiency of the steam = 67 ÷ 1,109, or 0·0604—that is, 6 per cent.

At 150 lbs. pressure the heat available is 1,118° — 1,042°, or 76°.

The efficiency of the steam = 76 ÷ 1,118, or 0·068—that is, 6·8 per cent.

At 200 lbs. pressure the heat available is 1,124° — 1,042°, or 82°.

The efficiency of the steam = 82 ÷ 1,124, or 0·073—that is, 7·3 per cent.

At 250 lbs. pressure the available heat is 1,130° — 1,042°, or 88°.

The efficiency of the steam = 88 ÷ 1,130, or 0·078—that is, 7·8 per cent.

In actual practice steam of 200 lbs. pressure enters the engine at a temperature of 381° F., and leaves it at the condenser at about 120°, so that it gives up in the engine 261°, which multiplied by 772 and divided by 33,000 is 6·1 H.P.

The triple-expansion engine of the mercantile marine consumes 16 lbs. per horse-power-hour, or 0·267 per minute. In such an engine a pound of steam produces 3·75 I.H.P.

Hence the steam efficiency of the engine is 3·75 ÷ 6·1, or 0·615.

An engine using steam at 250 lbs. pressure absolute receives it at a temperature of 401°, and rejects at 120°, giving up to the engine 281°, which is equivalent to 6·58 H.P.

A quadruple engine under these conditions consumes 14·5 lbs. of steam per I.H.P.-hour, or 0·242 lb. per minute, consequently a pound of steam produces in it 4·13 I.H.P.

Hence its steam efficiency is 4·13 ÷ 6·58, or 0·628.

A turbine using steam of 200 lbs. pressure and rejecting at 81° will absorb 300°, equivalent to 7·07 H.P.

Assuming it to consume only 12 lbs. of steam, the power generated in by a pound of steam is 4·99 H.P.

The efficiency of this turbine is 4·99 ÷ 7·07, or 0·706.

The Steam Efficiency of the Best Turbines is as high as 72 per cent. when of large size, and under favourable circumstances; that of turbines of good make and over 2,000 shaft horse-power can be taken at 64 to 66 per cent. With superheated steam those on shore of large size can be depended on to show an efficiency of 72 to 75 per cent.

The Steam Efficiency of Reciprocators of large size, good design, and good construction, 60 to 63 per cent. is satisfactory.

The following table gives the maximum amounts of work generated theoretically by a pound of steam during admission and expansion :—

TABLE XXVIII.—MAXIMUM WORK DONE BY 1 LB. OF STEAM PRESS. p_1, EXPANDING ADIABATICALLY TO PRESS. p_2, AND EXHAUSTING TO CONDENSER AT 1 LB. PRESS.

Initial pressure,	100 lbs.		125 lbs.		150 lbs.		175 lbs.		200 lbs.		250 lbs.	
Terminal p_2.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.	B.T.U.	H.P.
1·0 lb.	288·0	6·80	303·0	7·14	314·5	7·42	324·0	7·64	332·5	7·84	348·0	8·21
2·0 lbs.	286·8	6·76	301·8	7·09	313·3	7·26	322·4	7·61	331·3	7·81	346·8	8·18
3·0 „	277·2	6·54	292·6	6·90	309·2	7·17	313·5	7·40	322·2	7·60	337·2	7·95
4·0 „	266·2	6·28	281·9	6·65	294·0	6·93	303·2	7·15	312·0	7·36	328·2	7·74
5·0 „	258·0	6·09	273·5	6·45	285·0	6·72	295·0	6·96	304·0	7·17	320·0	7·55
6·0 „	249·9	5·90	266·9	6·27	278·0	6·55	287·9	6·80	297·4	7·01	313·9	7·40
7·0 „	243·9	5·75	259·3	6·11	271·9	6·42	281·8	6·64	290·8	6·86	307·8	7·26
8·0 „	237·5	5·61	253·5	5·96	265·5	6·26	276·8	6·53	285·5	6·73	302·5	7·13
9·0 „	232·0	5·47	248·0	5·85	260·5	6·14	271·3	6·40	280·7	6·62	297·0	7·00
10 „	226·9	5·35	242·1	5·71	254·0	6·00	265·5	6·26	275·9	6·51	292·1	6·89
11 „	222·1	5·24	237·1	5·59	249·8	5·89	260·7	6·15	270·5	6·38	287·1	6·77
12 „	217·0	5·12	233·0	5·49	245·5	5·80	256·1	6·04	265·4	6·26	283·0	6·67
13 „	212·2	5·00	228·6	5·39	240·7	5·69	251·6	5·93	261·1	6·16	278·7	6·57
14 „	207·4	4·90	224·1	5·28	236·4	5·60	247·2	5·83	257·3	6·07	274·4	6·47
15 „	204·0	4·81	220·0	5·19	232·0	5·47	243·5	5·74	253·0	5·97	270·0	6·37
16 „	200·6	4·73	216·6	5·11	228·6	5·39	239·6	5·64	249·9	5·89	266·4	6·28
17 „	197·4	4·65	213·3	5·03	225·0	5·30	236·0	5·56	246·5	5·81	262·6	6·19
18 „	194·3	4·58	210·1	4·95	220·0	5·23	232·9	5·49	243·3	5·74	259·9	6·11
19 „	191·3	4·50	207·2	4·88	219·4	5·17	230·0	5·42	240·5	5·67	257·3	6·06
20 „	188·4	4·44	204·4	4·82	217·3	5·12	228·0	5·38	238·0	5·61	255·4	6·02
25 „	176·0	4·15	191·3	4·51	205·0	4·83	215·9	5·09	225·4	5·31	243·0	5·73
30 „	164·8	3·89	180·0	4·24	194·5	4·59	205·0	4·83	214·9	5·07	232·0	5·47
35 „	155·2	3·66	169·6	4·00	184·2	4·34	195·2	4·60	205·2	4·84	222·2	5·24
40 „	146·2	3·45	160·5	3·80	175·1	4·13	186·0	4·40	196·0	4·62	213·0	5·02

Steam Efficiency may also be ascertained by referring to Table XXVIII., where it will be seen that a pound of steam expanding adiabatically from 200 lbs. absolute to 1 lb., when it enters the condenser, can theoretically give 332·5 B.T.U., equivalent to 7·84 horse-power.

A Turbine practically works on these conditions with a steam consumption of about 11·5 lbs., or 1 lb. of steam is good for 5·24 H.P. In this case, then—

The efficiency = 5·24 ÷ 7·84, or 66·8 per cent.

A triple-compound engine using steam at 200 lbs. pressure absolute expanding to 8 lbs. (nominal expansion rate 25), and exhausting at 1 lb. to the condenser, requires 15 lbs. of steam per H.P.-hour, so that 1 lb. is good for 4 H.P., and theoretically under these conditions develop 285·5 B.T.U., equivalent to 6·73 H.P. (*v.* Table XXVIII.). In this case—

(*a*) Efficiency = 4 ÷ 6·73, or 0·594, or 59·4 per cent.

But 1 lb. of such steam if expanded to its full extent is shown to be good for 7·84 H.P., then—

(*b*) Efficiency = 4 ÷ 7·84, or 0·51, or 51·0 per cent. only.

A Quadruple Engine working with steam at 250 lbs. absolute, expanding to 8 lbs., or nominally 31 times, exhausts to the condenser at 1 lb., consumes 14 lbs. of steam per H.P.-hour, so that 1 lb. is good for 4·286 H.P. But 1 lb. of steam is theoretically good for 302·5 B.T.U. or 7·13 H.P. under these conditions—

Then efficiency = 4·286 ÷ 7·13 = 0·601, or 60·1 per cent.

Taking the full value of 1 lb. of such steam as 8·21 (*v.* Table XXVIII.)—

(*b*) Efficiency = 4·286 ÷ 8·21 = 0·52—that is, 52·0 per cent. only.

High Pressure: its Advantages and Disadvantages.— Before the introduction of the compound engine for marine purposes, the boiler pressure had been as high as 60 lbs. in quite large steamers, in H.M. service, with the non-condensing engines of 200 N.H.P.; some of these were fitted in certain battle-ships during the 1855 Russian war; but after the compound engine secured the confidence of all classes of steamship owners, that pressure was very much exceeded with beneficial results.

Prior to the general use of the triple-compound engine 90 lbs. was a very common boiler pressure, and many ships' boilers were made for a working pressure of 100 lbs., and a few for as high as 110 lbs. The triple-compound engine itself was for a long time worked with steam of 150 lbs.; then 165 lbs. became a fashionable pressure, to be soon superseded by 175 lbs.; and now even 200 lbs. is general, although the economic gain with triples by going from 150 to 200 lbs. is questioned by some engineers who have made careful observations of all the conditions involved in the change.*

The objection to still higher pressures is rather of a practical nature, but can be safely overcome, since steam superheated to 600° F. may now be used. Steam at a pressure of 250 lbs. absolute has a temperature of 401° F., or nearly that of the melting-point of tin. It will, therefore, affect the condition of some of the metals with which it comes in contact, rendering the surfaces brittle and in a bad condition to withstand severe rubbing. Moreover, common unguents are vaporised, and the walls of the cylinders become too hot to condense their vapour when exposed to very high temperatures; but the heavy mineral oils now used, however, have a boiling point of

* The N.E. Coast Inst. E. and S. Standard Specification of 1917 gives 180 lbs. for cargo steamers with triples.

700° F. The difference in expansion of different metals is so considerable, that the utmost care must be exercised in the design and manufacture of the cylinder, etc., to prevent racking, which causes leakages and breakages. The pressure necessitates considerable thickness in all cast-iron work; and neglecting all considerations of weight or cost, this alone constitutes a source of objection and danger, inasmuch as the sudden exposure of thick masses of this metal to high temperatures on one side only is sure to distort, and very likely to fracture it. The liability to leakage is, of course, greater with the higher pressure, independent of temperature, and the danger to the attendants, from even small explosions, is very much increased. Again, in order to expand steam at this pressure, so as to obtain from it the maximum efficiency, it will be found necessary to use a series of cylinders; and although, as will be shown later on, the engine with four stages is not without virtue, a larger number will not commend itself to engineers generally for engines of the smallest and largest sizes. In practice, it has been found that an engine can be worked with steam of a pressure less than that usual, but superheated to the temperature corresponding to the higher pressure, and yet be more economic and efficient than a similar one working with that higher pressure of steam.

Efficiency of the Engine as a Machine.—The marine engine suffers loss, in common with all machines, from certain physical causes beyond the *absolute* control of the most skilful designer, and engineers can only aim at mitigating the evil, without entirely overcoming it. The chief cause of loss of energy is, of course, friction (1) of the piston, (2) of the stuffing-boxes, (3) of the guides and slides, (4) of the shaft journals, (5) of the valves and valve-motion. Another source of loss is that from the resistance of the pumps; and, finally, the inertia of the moving parts, which have a reciprocal action, as in the piston and rods, may be the cause of further waste of energy. Unless the momentum is balanced, or the energy imparted at one part of the stroke and stored in the heavy moving masses, is given out wholly by the end of the stroke, a serious loss ensues, and the mechanism has to sustain the strain of forces which might be otherwise usefully employed.

1. **The Friction of the Piston** in the vertical engine in smooth water depends on the pressure of the packing on the sides of the cylinder; so that, if the piston were solid and simply a good fit, there would theoretically be no friction, and in practice none beyond that due to the viscidity of the unguent and to the pressure on the sides of the cylinder from the rolling and general motion of the ship. This is what is required in a piston, and that one is most nearly perfect which is capable of moving steam-tight in the cylinder with least pressure of the packing, and so approximates to the condition of a solid one. Resistance due to this cause has been reduced to a minimum in modern engines by care in manufacture and skill in design; the cylinder is now truly and smoothly bored from end to end, the metal, which should be hard and close-grained, soon becomes polished and glazed, and in the best possible condition for smooth working; the packings of the piston are metallic, and the methods of pressing out the packing ring such as to ensure a uniform and evenly spread pressure of small magnitude. The loss from packing the pistons too tightly may become very great, and too much care cannot be exercised in attending to this most important part of the engine.

In horizontal and diagonal engines, the *weight* of the piston pressing on the side of the cylinder sets up friction, and as the cylinder wears in consequence more on the bottom than the top, it gets out of shape, thus necessitating more pressure on the packing-ring to maintain steam-tight contact, and thereby increasing the friction. The arrangements necessary for carrying the weight of the piston prevent all the better forms of piston rings and springs from being adopted in horizontal engines; so that the friction of pistons alone renders the engine less efficient than the vertical form. The most important improvement effected of late years, tending to the better working of diagonal engines, has been making the piston solid and of steel (*v.* fig. 90); in this way the weight has been considerably reduced, and the strength increased very materially; the *prevention* by this plan is better than the *cures* attempted by guide-rods, etc.

2. **The Friction of Stuffing-Boxes.**—Since the use of the higher pressures of steam this has become a very important factor in the consideration of engine efficiency. It may be extremely variable, as it depends so much on the care and judgment of the attendants, that, however carefully these parts may have been designed and constructed, their good working is entirely beyond the control of the designer. The manufacture of good and reliable metallic packing at reasonable rates has removed to a large extent this difficulty, so that now the glands of the high-pressure and medium-pressure cylinders are almost invariably fitted with some satisfactory form of metallic packing; and even those of the low-pressure cylinder are so treated in all important steamers, especially those of large size, although some engineers still prefer good vegetable packing for them. That the resistance may be very considerable is proved by the fact that the old trunk engines could be slowed down and nearly stopped by tightening the trunk glands, and in the ordinary piston-rod engine the speed may be appreciably retarded by the same process. The glands should never be so tight that the rods are rubbed absolutely dry in passing through them. For the efficient working of the engine it is better that a faint leakage of vapour should pass out with the rod, as that is generally an indication that the packing is just tight enough, besides which the moist vapour lubricates the packing and keeps it soft when of vegetable composition.

3. **The Friction of the Guides and Slides** is now, perhaps, the least important in the everyday experience of the losses to which an engine is liable; as those parts may be said to design themselves, and are now of ample surface, and they are in such a position as to command the attention of the engineer; they are also, as a rule, easily lubricated. In most classes of engines, the piston-rod guide is the chief one for consideration, and since, from the form of the rod-end, it is nearly impossible to give too small a surface to the *shoe*, it is seldom found, even in ill-designed engines, to give much trouble; but in certain forms of engine this is not always the case, so that care has to be taken both in designing and attending to the guides. The maximum pressure on the piston-rod guide of a marine steam engine is usually from two to three-tenths of the load on the piston, and supposing the ratio of maximum to mean pressure to be 1·50, and the coefficient of friction, under good circumstances, probably not less than 0·05, then

$$\text{Resistance of guide} = 0{\cdot}01 \text{ to } 0{\cdot}015 \times \frac{\text{load on piston}}{1{\cdot}50},$$

which means that from two-thirds to one per cent. of the energy of the piston is of necessity consumed in overcoming the friction of this one guide.

Improvements in the manufacture of, and choice of suitable metals for guides, have very sensibly diminished the loss from this cause; besides which the pressure on the guides has been in modern engines reduced by making the connecting-rod longer in proportion to the stroke. When cast-iron has become, by rubbing, *glazed* on the surface, there is no material better for guides. However, as some engineers will not wait for, or do not trust to, this state of metallic surface, but prefer the rubbing surface to be of a softer nature than the rubbed, it is not unusual to find white metal fitted to the "shoes." That some of the older engines were inefficient from loss at the guides, is proved by the rapid wearing of the old bronze shoes; the work necessary to convert so many cubic inches of metal into powder being the measure of avoidable loss at that point.

4. **The Loss from Friction at the Shaft Journals, etc.,** may be also very considerable, as the load on the piston is transmitted from the crank-pin to them, in addition to that caused by the weight of the shaft itself and the connections. The same may be said of the crank-pins, which have pressing on them the whole of that load, in addition to the weight of the rods. This friction may be very severe, especially in fast-running engines. It was formerly held that *friction was independent of velocity*, so far as movement through a fixed distance is concerned; that is, if a body be moved through 10 feet, the friction is the same if the movement takes place in 1 second or in 10 seconds; but if time be taken into account, the friction of moving the body ten times over the 10 feet in 10 seconds is ten times that of moving it *once* in 10 seconds. Since M. Morin made his experiments further investigations have been most carefully made by Mr. Tower and others, which show that friction does vary with the velocity, probably as the square root of the velocity; hence for modern piston speeds a larger allowance of surface is required than formerly. In a marine screw engine making seventy revolutions per minute, the friction is at least seventy times that of one revolution; and, consequently, if a paddle engine, having the same size of cylinders, and working with the same pressure of steam, makes only thirty-five revolutions per minute, its friction of journals will be half that of the screw engine per minute. As the first screw engines working without gearing were generally designed by men whose experience had been gained with the slower working paddle engines, it is not astonishing to find that the bearings were not always sufficient for the work on them, and perhaps the speed of the rubbing surfaces prevented the lubrication from being so efficient as had been the case previously, and so aggravated the evil. Again, the old paddle engine and geared screw engine had cylinders of longer stroke compared with their diameters than had the direct-working screw engine, and as the diameter of the shaft depends on the area of piston and length of stroke combined, while the pressure on the bearings is affected only by area of the piston, the diameter of the shaft might remain the same, although the size of the piston had been very much increased. Now, since most of the old rules for length of journals took cognisance of the diameter of shaft only, although the pressure on the journals might have been doubled, there was only the same surface to take it.

For example, a paddle engine of 5 feet stroke might have the same diameter of shaft as a screw engine of 2 feet 6 inches stroke, each having the same cylinder capacity; but the engine with the short stroke would have a piston area twice that of the long stroke, and consequently with the same steam pressure there would be double the load on the journals, and this with, generally, double the number of revolutions of the shaft.

Friction and Surface.—It was formerly an axiom that friction was independent of surface; but there must always have been limitations to such a statement. It meant that so long as the moving body was supported on a surface sufficiently large to prevent the unguent from being squeezed out, the resistance due to friction was the same fraction of its weight, whatever the area. Practical engineers, however, never had unbounded belief in this axiom. It is now clearly understood that when two surfaces are separated by a stratum of oil, or other unguent, they do not touch one another, and, therefore, each is sliding on the unguent. It is also generally supposed that the particles of unguent assume a globular form in the process, so that they form a kind of ball-bearing surface. Anyway, so long as the unguent is maintained in position, the resistance is small and is constant. It is, however, well known that when the same unguent is used with different metals having apparently equally smooth surfaces, there is a difference in the *coefficient of friction*, as the fraction is called; and, further, that certain metals will not work at all well on others—*e.g.*, soft steel on bronze. It is evident, then, that either the metals affect the unguent, or they do not really present equally good surfaces. It is known that oil acts chemically on copper, and, therefore, on copper compounds, while it has no action on tin and antimony. This may possibly account for the different behaviour of white metal and bronze.

Modern experiments have shown that surface has its influence, even within the safe limits; that the coefficient is higher with very light loads per square inch than heavier ones, so that when an engine is running quite light, the percentage of loss from friction is higher than when running full load. In the case of a marine engine running slow, the pressure per square inch on the guides, etc., will be much less in consequence of reduced load; so that the coefficient of friction will be, from *that* cause, somewhat higher. On the other hand, as the engine is moving slower, and inasmuch as friction varies as $\sqrt{V}$, the coefficient should be less, and so one may balance the other.

Tower found that at 90° F.,

$$\text{Coefficient of friction} = 20c\frac{\sqrt{V}}{P}.$$

Where V is the rubbing velocity in feet per minute, P the nominal pressure in lbs. per square inch, c a coefficient depending on the lubricant, which, for sperm oil, is ·0014, rape oil ·0015, mineral oil ·0018, and olive oil ·0019, the oil supply being liberal and constant.

For example, the coefficient for a guide on which the pressure is, say, 100 lbs. per square inch, the piston speed 900, and on which mineral oil is freely used,

$$F = 20 \times {\cdot}0018\,\frac{\sqrt{900}}{100} = {\cdot}0108.$$

If the oil is only sparingly supplied, as with a syphon lubricator, the coefficient may be four times this, or ·0432, which is very nearly what Morin stated to be the best result with well lubricated surfaces.

Temperature also affects the friction; the coefficient is reduced when the temperature rises so as to render the unguent fluid and not sticky. On the other hand, further rises of temperature tend to make it too fluid and thin. With some mineral oils, raising the temperature from 60° to 120° caused the friction to be quadrupled.

The aim of the engineer must be, therefore, to prevent metallic surfaces from coming into *actual contact*, for then the friction would be very severe, and soon cause the surfaces to abrade, and even, as is seen sometimes in the case of cast iron, to strike fire. The lubricant should be introduced at the points where the pressure is least, in order that those where greatest may be well lubricated. Friction at the journals, too, must always be a source of anxiety, though not nearly so much so now as formerly. Crank-shafts are more truly turned, although even now there is room for improvement; the foundations of the engines are more stiffly made, so that there is no springing at the bearings; the bearing surface is *pro rata* larger, and white metal is universally fitted to both crank-pins and main bearings.

5. **The Friction of Valve Motions** and of the valves is very considerable at all times, and may be severe when the valves are running dry. Even when the pressure on flat valves is partly relieved by frames, etc., on their back, the load on the rods was sometimes so great as to bend them. With the large increase of piston speed of necessity have come larger valves. This, with increased boiler pressure, forced most makers of marine engines to revive the piston valve for the high-pressure cylinders, and latterly to fit them to the medium-pressure and even to the L.P. cylinders. On the other hand, a few eminent builders stuck to flat valves for all their cylinders, and that, too, with a success that was somewhat remarkable, but it was with lower pressures and slower rates of revolution than now obtain.

Attempts have been made from time to time to use Corliss and other quick-shutting "drop" valves instead of slide valves. So far the results have been disappointing, and the increased efficiency due to quick cut-off and small clearance which such valves and gear give to land engines is now not yet attainable to the same extent with marine compound engines. But the marine engine may have its efficiency very much improved by some other form of valve which, while giving plenty of egress for the steam to the condenser, very much reduces the clearance now obtaining in low-pressure cylinders.

6. **Loss from the Pumps.**—In all engines, whether condensing or non-condensing, there is a loss of efficiency from the feed-pump when it is worked by the engine. It is true that the work done by it in forcing the water into the boiler is stored energy, and, therefore, not lost, but its low efficiency is a source of loss. It is now, however, the common practice for the feed-pump to be worked by an independent engine. This is convenient, and has advantages which will be discussed elsewhere, but it must not be overlooked that the cost in steam is greater than by the old method, as the independent engine consumes at least double the amount of steam per I.H.P. that the main engines do.

To some extent the air-pump may be said to store energy, for it takes water from under a pressure of, say, 2 lbs. per square inch, and places it at the disposal of the feed-pumps under a pressure of 15 lbs. The chief work of the air pump is to withdraw air and other gases from the condenser at the pressure in it, and deliver them at atmospheric pressure. This, except as a means to an end, is all lost energy, while the friction, etc., of this pump adds to the loss, and so reduces the efficiency of the engine.

Air-pumps, especially in warships, are sometimes worked by an independent engine (*v.* figs. 124 and 127); the same objections apply to this pump being dealt with in this way as to the feed-pumps, and there is not the corresponding advantage in practice, although there might be if the pump were regulated to run exactly to the requirements of the condenser.

From carefully made experiments with land engines, there is good grounds for estimating the power absorbed in driving the air pumps of a marine as varying from 5 per cent. of the gross I.H.P. in small engines to about 1 per cent. in large ones. If the air pump does not exceed one-twentieth the L.P. cylinder, it will be only from 2·0 to 0·7 per cent.

In an engine of 1,500 I.H.P. the air pump will absorb 3 per cent. of the power developed, while in one of 3,000 I.H.P. it will be 1·35 per cent., and in one of 5,000 I.H.P. 0·82 per cent.

The circulating-pumps render nothing for the large amount of energy put into them, and, viewed in this light only, would seem to be a great impediment to the engine. As a means to a most valuable end it is otherwise appraised; as a necessary adjunct to the surface-condenser it has conduced very much to the economy of the modern marine engine. Besides all the energy lost in the driving of this pump the greatest waste of the steam engine is taken by it, for its water conveys away all the latent heat required to make the steam. It has been suggested that by turning the stream of water towards the stern some propelling effect would be obtained, and some of this loss be prevented. So far no engineer has gone out of his way to effect this small economy.

The power absorbed in circulating the water through the condenser in steam engine installations on land is heavy; much more so than that on board ship, where the condenser itself is commonly below the water-level, and there is no "head" but that due to resistance in tubes, etc., to be pumped against. The variation found in these land installations is, however, instructive, for in cases of 1,500 I.H.P. the circulating pumps absorb as much as 10 per cent., at 3,000 I.H.P. stations it is 4·5 per cent., and at those of 5,000 I.H.P., 3·2 per cent. On shipboard, with the discharge valve at or below the water line, 1·7 to 2·5 per cent. of the main engine power is sufficient in the tropics, and about a half in temperate zones. In merchant ships, with the discharge well above the water line, the loss is 2·2 to 3·0 per cent. in the tropics.

7. **Inertia of Moving Parts.**—There should be no losses from this cause if proper provisions are made, for all the energy stored in them in getting them to their maximum motion should be usefully employed and absorbed before coming to rest. In the older engines, as in many modern ones, considerable loss arose from the fact that much of the energy stored in the pistons and other heavy moving parts was not usefully employed, but was wasted in vibrating the engine itself and the ship. This, however, is fully dealt with in Chap. xxix. There will, of course, still be small losses due to friction on bearings and guides from the inertia of the moving parts after the engine has been balanced, for the balancing cannot bring all the forces into one plane.

It will be seen by the foregoing that the energy lost in a marine engine is not a definitely fixed quantity, and is dependent neither on design nor on construction alone, but rather on the degree of care exercised by those having the charge of it. That much may be saved by good and careful designing and workmanship is evident, but, within certain limits, a good engine may prove less efficient than a really inferior one from mere lack of proper attention from those in charge, and especially if they are not supplied with good and suitable lubricants.

The Losses due partly to Mechanical Defects and partly to Physical Causes are those which cannot be classed as belonging to the engine as a machine, nor to it as a heat engine simply. The most important of these is consequent on the employment in its construction of materials having a high power for conductivity of heat. The steam pipes were generally of copper, and when of steel are comparatively thin, so that there is much loss of heat from their surfaces. Careful covering with material having a low conductivity does much to prevent this waste, but, with the high pressures now used and consequent high temperatures, unless the lagging is very thick and the pipe *flanges well covered* there is still great loss. The loss, too, is not limited to the mere heat which escapes, for if saturated steam—that is, steam containing as much water as it can—is robbed of any of its heat on its road to the cylinders it deposits some of its water; this water obstructs the free passage of steam through valves and passages, and, till forced through the escape valves or drain cocks, obstructs the pistons themselves. The efficiency of the engine as a heat engine is also materially affected by the presence of water in the cylinders. The losses are very manifest when the speed of the engine is, as often happens, materially checked by water coming with the steam. The gain from superheating the steam for the marine engine arose, and will again be found to arise, largely from the steam being quite dry rather than from the higher temperature at the cylinder.

Liquefaction takes place on expansion in the cylinders in spite of steam jackets and superheating, and a small amount of water to lubricate the internal parts is a good thing, especially as the use of oils is very restricted or wholly done away with nowadays. Notwithstanding, the cylinders should be as carefully "lagged" as the steam pipes, and the covers and flanges of the high-pressure cylinder should be as carefully covered as any other part.

It is very doubtful if steam, during expansion in the fast-running marine engine, can take any appreciable heat from the jackets; that such jackets add somewhat to the efficiency of the engine must, however, be admitted when they are properly drained; it is probable, therefore, that the gain is one of mechanical efficiency, from the fact that the cylinders are freer of water with steam jackets in use.

If this argument is based on fact, a jacket to the low-pressure cylinder might be of some use, and even outweigh the loss arising from the reheating of steam about to enter the condenser at exhaust.

Superheating the steam so that it might continue wholly in the vapour state till it reaches the condenser has been the hope of some sanguine inventors. Superheating the steam with heat that otherwise is lost is obviously the most desirable thing as a means of economy. Superheating, however, must have its limits, for in economising steam there may be great loss of energy, as, for example, by excessive friction of the high-pressure piston, valves, and rods from want of a lubricant when the steam is so dry as to absorb it all in the vaporous state. As a matter of fact the steam may leave the marine boiler with a large amount of superheat, and even on reaching the cylinder still have a fair margin, but the piston will not have moved many inches before it is saturated steam. If water deposit in the high-pressure cylinder is to be avoided, the superheater will require to have special heat supply or its own fire, and unless very special means are adopted for lubricating the piston and valve will work badly and cut the surfaces.

Superheating went out of fashion on the advent of steam of 60 lbs. pressure, partly due to the fact that its temperature, 307° F., was nearly as high as superheating had then gone, and to the restrictions put by the Board of Trade on superheaters and their somewhat rapid decay.

Metallic packing was unknown at that time, and the gain in economy by the compound engine over the old surface condensing 30-lb. pressure ones satisfied fully the enterprise of the good folks in those days of good freights and cheap fuel.

By superheating the steam for electric light installations on shore great reductions in consumption of fuel have been effected (fig. 64). No doubt similar results may also be obtained by employing satisfactory superheating apparatus on board ship, as shown below, but great care must be taken in the design and construction of the piston and valves of the first cylinder into which such steam enters, or the loss may exceed the gain in steam efficiency.

Experiments with Superheated Steam in Modern Times* have been made, and the results of some very interesting ones were given by Mr. Felix Godard in a paper read at the Inst. Naval Architects. These trials were made with the screw steamers "Garonne" and "Rance," each 300 feet long, 40 feet beam, and 25·5 feet deep. Mean draft of water 21 feet, gross register 2,700 tons.

The "Garonne" has no superheater, but her total heating surface is equal to that of the "Rance," together with the surface in that ship's superheater. That is, the total surface exposed to heat is the same in each ship. Their engines have cylinders 23 inches, 36 inches, and 59 inches diameter and 42 inches stroke.

	"Garonne."	"Rance."
Boiler pressure,	178 lbs.	177 lbs.
Steam temperature,	378° F.	518° F.
Heating surface of boilers,	3,767 sq. ft.	2,982 sq. ft.
Superheating surface of boilers,	none	785 sq. ft.
Revolutions per minute,	72·3	75·4
Indicated horse-power,	1,104	1,304
Coal consumed per I.H.P. per hour,	1·12 lbs.	0·90 lb.
Coal per mile on service,	154 lbs.	126 lbs.
Increase in power,	..	18·1 per cent.
Decrease consumption per I.H.P.,	..	20·1 ,,
Saving of fuel on voyage,	..	18·2 ,,

Two other ships were fitted in the same way—that is, one, the

* *Vide* Appendix E for more recent experiences.

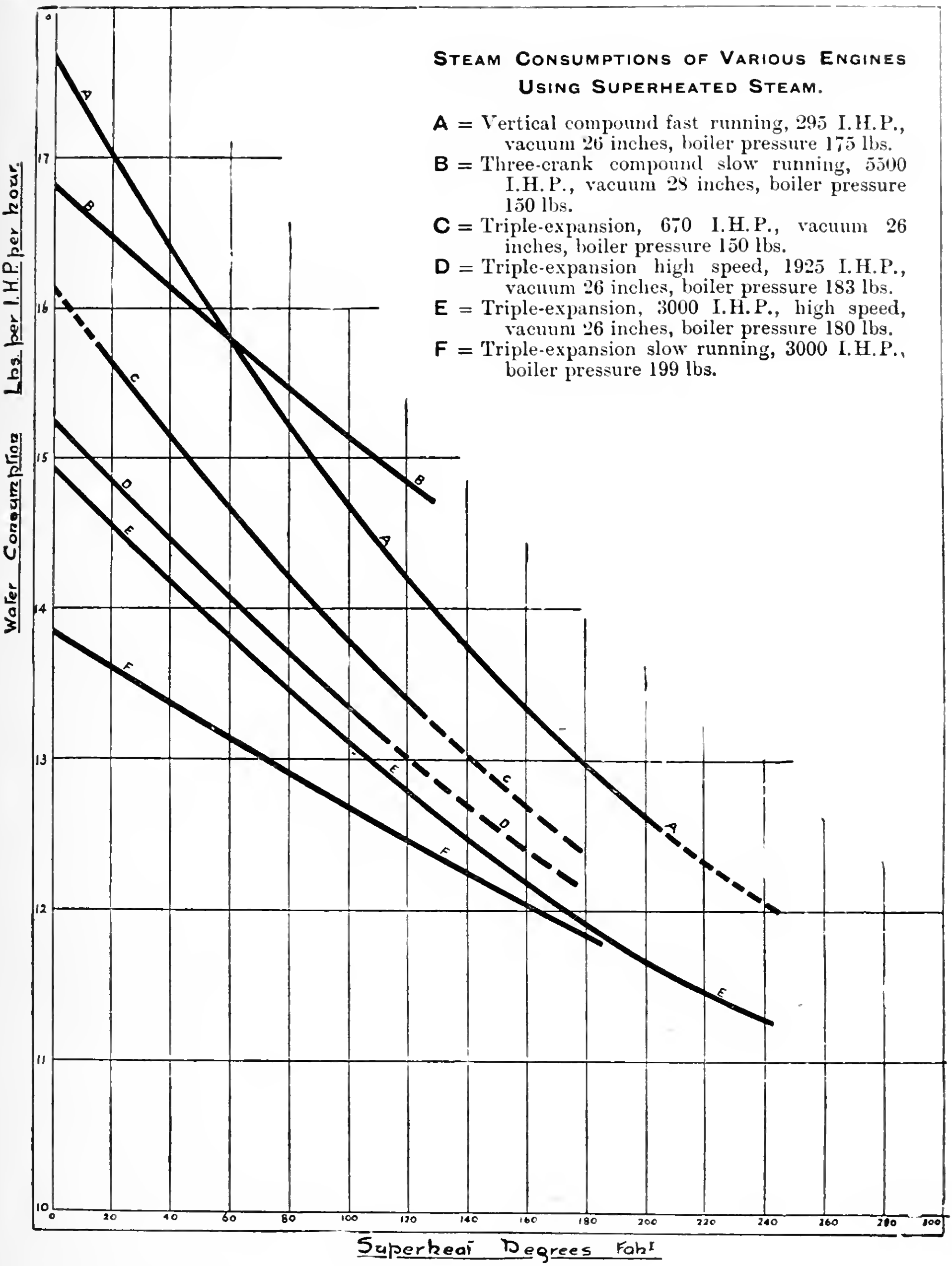

Steam Consumptions of Various Engines Using Superheated Steam.
A = Vertical compound fast running, 295 I.H.P., vacuum 26 inches, boiler pressure 175 lbs.
B = Three-crank compound slow running, 5500 I.H.P., vacuum 28 inches, boiler pressure 150 lbs.
C = Triple-expansion, 670 I.H.P., vacuum 26 inches, boiler pressure 150 lbs.
D = Triple-expansion high speed, 1925 I.H.P., vacuum 26 inches, boiler pressure 183 lbs.
E = Triple-expansion, 3000 I.H.P., high speed, vacuum 26 inches, boiler pressure 180 lbs.
F = Triple-expansion slow running, 3000 I.H.P., boiler pressure 199 lbs.
Water Consumption Lbs. per I.H.P. per hour.
Superheat Degrees Fahr.
A
B
C
D
E
F
10
11
12
13
14
15
16
17
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300

Fig. 64.

"Guadaloupe," had the same quantity of heating surface, 13,509 square feet, as the heating and superheating surface together of the other, the "Peron." The temperatures were 378° as before, and only 460° with the superheaters; the indicated horse-power was practically 6,585 and 6,750 with 88 revolutions, the speed of the "Peron" being on service 16·95, against the 16·6 of the "Guadaloupe." These ships were 430 feet long, and of 6,800 tons gross register.

Another source of loss is the resistance due to the friction of the steam in passing through the pipes, passages, and valves; and although here again there is not a total loss, still it is not compensated for in the way that the engineer desires. As friction causes heat, so the friction of the steam along the surface of the pipes and passages generates heat; but since this heat is not allowed to escape, it is taken up by the steam, and tends to superheat it. The loss due to this cause in the steam-pipes is probably very small, especially when the pipes are of such a size that the velocity of steam through them is not excessive, and Mr. D. K. Clark found that it is inappreciable when the velocity is not more than 130 feet per second with very dry steam, and 100 feet per second with ordinary dry steam. The greatest loss is when the steam has to pass through narrow orifices where the perimeter of such orifices is large compared with the area, as is the case at the steam ports of a cylinder; this is called "wire-drawing" the steam, and there is always a loss of pressure from this cause, even when the area through which the steam passes is equal to that of the section of the pipe through which it has previously passed. When the area is reduced, and the perimeter is large, the loss is, of course, still greater, and hence the loss at the ports of a cylinder, where the cut-off is early by means of a common slide-valve, is very considerable, and may amount to as much as 10 per cent. of the pressure, unless the ports are very large, or the travel of the valve exceptionally long. To obviate such an evil, the double and treble-ported valves are used, and other plans adopted, whereby increased area of opening to steam may be obtained. If care is taken so as to avoid all unnecessary obstructions to the passage of the steam in the pipes and stop-valves, and there be sufficient opening of the port at the beginning of the stroke, the loss of initial pressure should not exceed 2½ per cent., and in some marine engines there is no appreciable fall of pressure from the boiler to the cylinders. There is also a loss of energy when the steam enters the cylinders due to sudden change in velocity, which will be from 150 feet to 15 feet per second in large engines, and even greater in small engines, where the piston velocity is very much less than 10 feet per second. This cannot be avoided in any way, as it is practically impossible to increase the piston velocity to even one-half that of the steam; and it would be excessively inconvenient to increase the area of ports, etc., so that the velocity of steam should more nearly approach that of the piston. But as the loss from this cause is very slight indeed, no extra cost expended in attempting to avoid it would meet with an adequate return.

A considerable amount of heat is lost by the radiation from those parts which are alternately exposed to the hot steam and to the atmosphere; and this was especially great in trunk engines, where the surface of the trunks is very large, and, being hollow, of course have the inner surface giving off heat as well constantly. Fortunately, the surfaces of the piston-rods, trunks, etc., soon become highly polished, and so do not radiate the heat so quickly as they would were they rough. This loss, too, cannot be avoided, or even reduced to any appreciable extent.

Finally, there is the loss due to the heat conducted from the cylinders, pipes, etc., to the other parts of the engine with which they are connected, and which pass it away by radiation at their surfaces.

CHAPTER VII.

ENGINES, SIMPLE AND COMPOUND.

Elementary Steam-Engine.—The steam-engine, in its most elementary form, has only one cylinder, into which the steam is admitted at each end alternately, so as to move the piston backwards and forwards, and having performed its work is then allowed to escape into the atmosphere. Although from certain causes this was not the form of steam-engine as first invented, it is nevertheless the most simple one, and by taking it as the origin, the genesis of the marine steam-engine can be better explained. As the engines for marine purposes are still largely those having cylinders and pistons, it will be unnecessary in this chapter to deal with any other forms.

Genesis of the Compound Engine.—The exhaust steam issuing from an engine having a late cut-off and an initial pressure of, say, two atmospheres (or 15 lbs. above atmospheric pressure), would attract the attention of an observant engineer from the force with which it emerges from the exhaust-pipe, and would naturally lead him to inquire how so great a waste of energy *might* be avoided. It would be clear to him that there was sufficient remaining in it to do useful duty after it had accomplished its work in the cylinder. Being acquainted with the steam-engine of Watt, he would suggest that, instead of allowing it to escape into the atmosphere, it might be conducted to the cylinder of a condensing engine, which could work with a steam pressure of one atmosphere, and while operating the piston of this second engine, would cause no more back pressure in the first cylinder than before. Such an arrangement would be a combination of a high-pressure and a condensing engine, and hence it was called a *compound engine.*

The engineer to-day would suggest that the steam be taken to a low-pressure turbine, as is often done, and the turbine exhaust to a condenser, whereby a better use would be made of it than the Watt engine did.

The idea of a compound engine, however, was due to the genius of Hornblower, a Cornish engineer, who in 1781 took out a patent, in which he claimed:—"I use two steam vessels, in which the steam is to act, and which in other steam-engines are called cylinders. I employ the steam, after it has acted in the first vessel, to operate a second time in the other, by permitting it to expand itself, which I do by connecting the vessels together, and forming proper channels and apertures whereby the steam shall, occasionally, go in and out of the said vessels, etc."

Arthur Woolf, in his patent taken out in 1804, states that "if the engine be constructed originally with the intention of adopting my said improvement, it ought to have two steam vessels of different dimensions, according to the temperature or the expansive force determined to be communicated to the steam made use of in working the engine; for the smaller steam vessel or cylinder must be the measure of the larger. . . . The small cylinder should have a communication, both at its top and bottom, with the boiler which supplies the steam. . . . The top of the small cylinder should have a communication with the bottom of the larger cylinder, and the bottom of the smaller one with the top of the larger, with proper means to open and shut those alternately by cocks or valves, etc., . . . and both top and bottom of the large cylinder should, while the engine is at work, communicate alternately with the condensing vessel." He proposed to use steam at a pressure of 40 lbs.

Expansive Engine.—If the observer, however, happened to be better acquainted with the expansive force of steam than with the use of a turbine and condenser, he would suggest that the steam should be cut off at such an earlier part of the stroke as would ensure its pressure, at emission, being only slightly above that of the atmosphere, and all available energy abstracted. Such an engine would naturally be called *expansive*, in contradistinction to the elementary engine, working without expansion. Any further attempt at increased expansion would prove less fruitful, as the steam, expanded below the pressure of the atmosphere, will fail to escape into it when opened to exhaust; besides which, the "back" pressure on the other side of the piston would, during the latter part of the stroke, be greater than the forward. Then it is that by connecting the exhaust pipe to the condenser, in which the pressure would be 10 or 12 lbs. below that of the atmosphere a higher rate of expansion, could be obtained.

Effects of Increase of Pressure.—If an engine is to work economically, so far as steam is concerned, it has been stated that the terminal pressure, before admission to the condenser, should be as low as possible consistent with good working.

It is now easy to maintain a vacuum of 28 inches in a modern condenser, but 25 inches was usual with the jet condensers. As some engineers may still prefer to work their engines with only 24 to 25 inches of vacuum (for the sake of obtaining warm feed-water), let it be assumed, for the sake of argument, that 24 inches is the vacuum in the condenser. When the full benefit of expansion is required, the terminal pressure should not exceed 7 lbs., which will be 3 lbs. above the back pressure. With a turbine expansion may go on till the difference is less than 1 lb.

In order to appreciate fully what is encountered in making advances in boiler pressures, it will be well to compare two engines working under the conditions set out above. Suppose these two engines to have each one cylinder of the same diameter and stroke, the boiler pressure of the first to be 2 atmospheres, or 30 lbs. absolute, that of the second 3 atmospheres, or 45 lbs. absolute, the terminal pressure in both cases to be 7 lbs., and the back pressure 4 lbs. The cut-off in the first will be $\frac{7}{30}$ of the stroke, or a rate of expansion of 4·285, and the mean pressure with an initial of 30 lbs. is 1·74 lbs.; deducting 4 lbs. of back pressure, the *effective* mean pressure will be 13·4 lbs. In the second engine, the cut-off will be $\frac{7}{45}$ of the stroke, and the rate of expansion 6·43, and the mean pressure with an initial of 45 lbs. is 20 lbs.; deducting, as before, 4 lbs. for back pressure, the *effective* mean pressure is 16 lbs. The *effective initial* pressures will be 30 — 4, or 26 lbs., and 45 — 4, or 41 lbs., respectively.

Since the cylinders are of the same capacity, and the terminal pressures are the same, each engine consumes the same *weight* of steam; but the total heat of evaporation of steam from the temperature corresponding to 4 lbs., and at that corresponding to 45 lbs., is 1,130° F., while at that corresponding to 30 lbs. it is 1,121° F., there will be, therefore, an expenditure of fuel to obtain the steam at 45 lbs. slightly in excess of that at 30 lbs. As this, however, amounts to less than 1 per cent., it may be neglected, and the cost of the steam assumed to be the same in both cases. It will be seen then that, with this advance of boiler pressure, there is an advance in mean pressure, and the gain in power amounts to nearly 20 per cent.; but the initial load on the piston of this more economic engine is 57 per cent. higher than that on the more wasteful one, and consequently the rods, framing, etc., must be 57 per cent. stronger; moreover, the shaft will be increased in size, and the cylinder and passages must be stronger. Altogether, then, the engine will become heavier and more costly, as the boiler pressure is increased, while the expenditure of fuel is less per horse-power.

To render the comparison strictly fair, it would be necessary to take two engines of *equal power*, so that if the stroke of the pistons is the same, their areas will be inversely proportioned to the mean pressures, and, consequently, the initial loads will now be 26 lbs. and $\frac{13 \cdot 4}{16}$ of 41, or 34·5 lbs., which gives an excess of 31·3 per cent.

Progress Made by Early Marine Engineers.—The early marine engineers, however, did not advance on these lines as a rule, for nearly the same rate of expansion was observed at full speed with steam of three atmospheres as had obtained with steam of atmospheric pressure; consequently very little benefit was derived in economy, compared with what might have been the case had they done differently. The chief result accruing from the increased boiler pressure was in practice the larger indicated horse-power developed by engines of the same size as formerly, partly due to the augmentation of mean pressures, and partly to the increased piston speed resulting from them.

Engines of as much as 200 N.H.P., working with steam of 60 lbs. pressure, were supplied to the Navy by Messrs. Penn and Messrs. Maudslay as early as 1853; but a period of more than fifteen years elapsed before the Admiralty again employed such pressure in any larger ship than a gun-boat.

By the following table a comparison can be made of four typical simple engines working with steam at different pressures, and their performance under the varying conditions:—

TABLE XXIX.

Load on safety valves,	lbs.	30	45	60	60
Diameter of cylinder,	ins.	50	50·8	51·1	38
Initial absolute pressure,	lbs.	45	60	75	75
Cut-off,	stroke	$\frac{6}{10}$	$\frac{3}{10}$	$\frac{2}{10}$	$\frac{6}{10}$
Back pressure,	lbs.	4	4	4	4
Mean effective pressure,	,,	36·77	35·66	35·15	63·95
Terminal ,,	,,	27·00	18·00	15·00	45·00
Maximum load on piston,	,,	80,483	113,456	145,550	80,514
Mean ,, ,,	,,	72,180	72,180	72,180	72,180
Ratio of max. to mean,	,,	1·115	1·572	2·016	1·115
Weight of steam used,	,,	893	609	515	868

Examples (1), (2), and (3) show conclusively that by increasing the boiler pressure and rate of expansion, so as to obtain nearly the same mean pressure, the weight of steam used is considerably diminished and the terminal pressure considerably reduced, but that there is an increase in the maximum load on the piston proportionate to the *absolute* pressure in the boiler, so that the ratio of maximum to mean pressure of example (3) exceeds that of example (1) by more than 80 per cent.

Although such an increase in weight of machinery as would be necessitated by so great an increase in load, may not be of much importance in some ships, in others it would be prohibitive; for when the power required for certain speeds of ship becomes large compared with the displacement, it requires the utmost care in design to keep down the weight, so as to admit of the engines being carried by the ship on the required draught of water. For this reason, in actual practice, it was found advisable to use steam of under 50 lbs. pressure (above the atmosphere) in very fast river steamers, or even in high-powered steamers for Channel service of moderate size, on account of the limited speed of piston obtainable with the paddle-wheel, until, by means of forced draught, constructing the ship and engines as much as possible of steel, and a special design of light compound engines, pressures of 100 to 125 lbs. could be employed in such ships; now in some few cases triple-expansion engines using steam of 175 lbs. pressure are fitted in paddle steamers of high speed.

Example (4) is given that the effect of two widely different boiler pressures may be compared, when the rate of expansion is the same. The mean pressure is, of course, much higher, and, but for the back pressure being constant, would bear the same proportion to that at the boiler pressure of 30 lbs. as the initial absolute pressures—viz., 5 to 3. The weight of steam used is very little less than that of 30 lbs. pressure, and, owing to the reduction in the size of the piston, the maximum load in this case is practically the same as in example (1). The pressure at exhaust is exceedingly high, being 45 lbs. absolute, or 30 lbs. above that of the atmosphere, so that it is capable of doing considerably more work, if admitted into another cylinder of larger size, than that of the first; and even if admitted into one of the same size (provided it finally exhausts into a condenser), more work will be obtained from the steam than if it is allowed simply to escape into the atmosphere.

Receiver Compound Engine.—Now, suppose that an engine working under the conditions set out in example (4) (so far as pressure and cut-off are concerned) exhausts into a steam-tight space, so that there is back pressure in front of the piston equal to the pressure in this receiver of the exhaust steam; and suppose, further, that the steam is taken away by another cylinder from the receiver at the same rate as it is supplied by the cylinder, there will then be a constant mean pressure maintained there. For the sake of fixing the application to example (4), suppose, again, the pressure in its receiver to be 30 lbs. *absolute*, then the mean pressure in the cylinder will be $67 \cdot 95 - 30 = 37 \cdot 95$ lbs. only. Now, suppose a second and larger cylinder to be supplied with steam from the receiver at such a rate that there is no change of pressure in it (this being accomplished by so arranging the cut-off in the second cylinder, that the weight of steam taken by it equals the weight of steam exhausted from the first one); the cut-off may be determined from the formula $p\,v =$ const.; so that if V be the volume of the second cylinder and v that of the first, 45 lbs. the terminal pressure in the small and 30 lbs. the initial in the large cylinder. Then, cut-off in the second cylinder $= \frac{45\,v}{30\,\mathrm{V}} = \frac{3}{2} \times \frac{v}{\mathrm{V}}$, and if the ratio of V to v is 3, the cut-off in the second cylinder is $\frac{1}{2}$ stroke, and the rate of expansion in it 2.

The mean pressure, with an initial pressure of 30 lbs. and a rate of expansion 2, is 25·38 lbs.; and allowing for a back pressure of 4 lbs., the mean effective pressure in the second cylinder is 21·38 lbs.

Since the area of the second piston is three times that of the first, the work done in the second cylinder is equivalent to what might be done by one of the same area as the first, with a mean effective pressure three times as great, or 64·14 lbs. per square inch. It will be seen from this that the total work done by the combined cylinders is the same as would be done by the original cylinder with a pressure of 37·95 + 64·14, or 102·09 lbs. per square inch; hence we find that there is a gain of nearly 60 per cent. by the introduction of the second cylinder. So far, the compound engine would be undoubtedly more economical than the expansive engine, as exemplified in examples (1), (2), and (4), but less so in this particular instance than example (3).

Expansive and Compound Engines Compared.—To examine the relative economy of a compound and of a simple expansive engine, it is necessary that they should both work with the same boiler pressure and the same rate of expansion. Now examples (3) and (4) satisfied the first condition, and if the second is also satisfied, then they may be compared. The rate of expansion in example (4) is 1·666, and since the volume of steam in the second cylinder at the end of its stroke will be three times that in the first at the same period, the total expansion effected by both cylinders will be $3 \times 1{\cdot}666$, or five times. The cut-off in example (3) was two-tenths the stroke, and therefore its rate of expansion is five; so that these two examples may be compared as to the efficiency of the steam. The effective pressure of the compound system may be referred to the large cylinder, in the same way in which it was referred to the small one, and will be that actually on the large cylinder, together with that on the small one *divided* by the ratio of their capacities; hence, effective mean pressure referred to the large cylinder is $\left(21{\cdot}38 + \frac{37{\cdot}95}{3}\right)$, or 34·03 lbs. per square inch. It will be seen that this is 1·12 lbs. less than that obtained in the simple expansive engine, and therefore a loss has occurred somewhere in the compound system.

Suppose, now, that the cut-off in the large cylinder is so altered that the pressure in the receiver is 45 lbs., so that it receives steam at the same pressure as that which exhausts from the small one; in this case there will be no "drop" in the pressure from commencement of exhaust to the end in the small cylinder.

The mean effective pressure in the small cylinder is now 67·95 — 45, or 22·95 lbs. per square inch.

The cut-off in the large cylinder $= \frac{45\,v}{45\,\mathrm{V}}$, or $\frac{1}{3}$ the stroke, which gives 3 as the rate of expansion.

With an initial pressure of 45 lbs. and rate of expansion 3, the mean pressure is 31·5 lbs.; allowing 4 lbs. for back pressure, the effective mean pressure is 27·5 lbs.

Referred to the large cylinder the effective mean pressure of the system is now $27{\cdot}5 + \frac{22{\cdot}95}{3}$, or 35·15 lbs. on the square inch, or *exactly the same* as that obtained in the simple expansive engine.

Effect of "Drop" in the Receiver.—It is seen from the above, then, when no "drop" occurs there is no loss of efficiency; but that when the pressure in the receiver is less than the terminal pressure in the small cylinder, there is somehow a loss of effective mean pressure. This arises from the steam being allowed to *expand* from the small cylinder into the receiver *without doing work.* But it is known that, when this takes place, the steam becomes somewhat superheated; for, inasmuch as the loss of pressure has occurred without conversion into external work or loss of heat in any other way, it must appear in some other form. Although this loss is not wholly recovered, it must be to some extent reduced by the benefit which the steam derives from the superheating in expanding in the large cylinder.

Division of the Work.—It will be also seen that, as the ratio of the cylinders' capacity is 3, and the effective mean pressures 22·95 lbs. and 27·5 lbs., the work done in the small cylinder to that in the large is as 22·95 to 27·5 × 3, or nearly 1 to 3·6; while in the former case it was as 37·95 to 21·38 × 3, or nearly 1 to 1·7.

Therefore, with an *earlier* cut-off in the large cylinder, *more work* is developed in it than is the case when with a later cut-off; moreover, with this ratio of cylinders, in order to get the highest efficiency of the steam, the ratio of the work done is as 1 to 3·6; and the initial pressure on the large piston is 41, and on the small $\frac{75 - 45}{3}$, or 10, as against 71 on the expansion engine of *equal* size; and even if the compound engine were arranged with one cylinder above the other, the combined initial pressure would be 41 + 10, or 51, as against 71 of the simple expansive.

Direct Expansion Compound Engine.—The compound engine may, however, work without any intermediate receiver, if the pistons are arranged to move simultaneously, either in the same or opposite directions. To consider this case—suppose the cylinders to be side by side, and the pistons to move in opposite directions, as originally proposed by Woolf, so that when the small piston has *receded* one-tenth of the stroke, the large one has advanced by exactly the same amount, and the space between them is $0·9\ v + 0·1\ V$; the volume of steam at commencement of exhaust is v, and the pressure at that period, as before, 45 lbs.; the volume at any point of the stroke $\left(\frac{n}{10}\right)$, or the space between the pistons

$$= \frac{n}{10} V + \frac{10 - n}{10} v = \frac{n}{10} (V - v) + v,$$

and since $V = 3\ v$, space between the piston at n-tenths of the stroke $= v \left(\frac{n}{5}\right) + 1.$

The pressure at this point $= 45 \div \left(\frac{n}{5} + 1\right)$. The pressures at every tenth of the stroke from 0 to 10 will be 45, 37·5, 32·14, 28·12, 25·0, 22·5, 20·45, 18·75, 17·3, 16·07, 15·0; the mean of which is 24·78 lbs. per square inch. Deduct this from 67·95, and the effective mean pressure in the small cylinder is 43·17; deduct 4 lbs. from 24·78 lbs., and the effective mean pressure in the large cylinder is 20·78 lbs.

The effective mean pressure of this system, referred to the large cylinder, is now $\left(20{\cdot}78 + \frac{43{\cdot}17}{3}\right)$, or 35·17 lbs. per square inch, which is the same as that obtained by direct expansion in one cylinder—example (3).

It will be observed, however, that the ratio of the power exerted by the small cylinder to that exerted by the large one, is as 43·17 to $3 \times 20{\cdot}78$, or 1 to 1·44, being nearer an equal distribution of the work than in either of the cases of the intermediate receiver engine. This latter result is caused principally by the decreased back pressure in the small cylinder.

In actual practice there are certain causes which materially modify the results shown by both of these forms of compound engine, as illustrated in the foregoing.

In the receiver compound engine, the pressure in the receiver is not constant, because of its limited size; the difference in the periods of exhaust and admission, and the cushioning after cut-off in the L.P. cylinder by the small piston cause considerable oscillation.

The direct expansion engine is only nominally without a receiver, as the space between the small piston and the large one is often necessarily considerable (*v.* fig. 75), from the size of the communication pipes and the valve-box of the large cylinder. The valve of the L.P. cylinder cuts off some time before the small one ceases to exhaust, causing cushioning in the latter and in the spaces; the small cylinder also commences to exhaust before the large one can take steam.

Direct expansion compound cylinders have, however, been arranged so that one cylinder communicates directly with the other, without any intervening space, by placing the cylinders side by side, and causing the pistons to operate on cranks set opposite one another (*v.* fig. 187). But such engines have, for other reasons, proved generally unsuitable for propelling.

Requisites in the Marine Engine.—A marine engine must be (1) when required readily started, stopped, and reversed; (2) it should have a turning moment or torsion as nearly uniform as possible; (3) it must be able to work *continuously* for long periods without stoppage from any cause; and (4) and lastly, it must be economical.

The first condition is a *sine quâ non*, and is generally fulfilled by having two or more cylinders operating on cranks at suitable angles.

The second condition is absolutely necessary when weight is a serious consideration, and is very fairly satisfied by the two or more cranks at proper angles, and the cylinders so designed as to divide the work nearly equally between them.

The third condition will depend very much on the *variation* in stress, so that the engine with small initial pressure in each cylinder compared with mean pressure, is more likely to fulfil it than one with large initial pressures.

The fourth condition, which is of the utmost importance to the shipowner, is very well complied with by the triple- and quadruple-compound engines in all sizes, and by the turbine and combined turbine and reciprocator in large sizes.

With two cylinders and cranks at right angles, there must inevitably be some amount of "drop," if the work is to be evenly divided when the power is as great as usually developed by a marine engine at service speed. The crank of the H.P. cylinder should lead—that is, should be in advance of the other crank by 90°, or such other angle as it is deemed best to set the cranks at. When this is the case, the small cylinder begins to exhaust just after the crank of the L.P. cylinder has got well over the centre, and tends to maintain a constant pressure on the large piston through the earlier portion of its stroke, and at cut-off the pressure in the receiver is not much below its average pressure. If, on the other hand, the crank of the large cylinder leads, exhaust takes place only a little before cut-off in the large cylinder, and causes a hump in the indicator-diagram, showing an increase in the amount of "drop," and that with no diminishing in the mean back pressure in the small cylinder. Engines having the low-pressure engine crank as the leading one were also generally unhandy.

The first triple-compound engines were, as a rule, designed so that the high-pressure crank "led," or was in advance of the medium-pressure crank by 120°, and the medium-pressure crank 120° in advance of the low-pressure crank; but in modern practice it is usual to fit the low-pressure crank in advance of the medium-pressure crank, etc., as in this way there is less variation in temperature in each cylinder, although the load on the pistons is less during the first half of the stroke and greater during the second half than is the case when the high-pressure crank leads, and the engine is not appreciably less "handy" (*v.* fig. 164).

The first cylinder of a compound system is called the "high" pressure, and the last the "low," from their association with the condensing and non-condensing engine. For convenience in speaking of them, they are designated by the initials H.P. and L.P. Hitherto, in the chapter, all comparisons of the compound with the simple expansive engine have been made on the supposition that the expansive engine has only one cylinder of the same capacity as the low-pressure one of the compound system. To render the comparison perfectly fair, it will be necessary to take such cases as may be found in actual practice. In triple-expansion engines the middle cylinder is called the "medium-pressure," and sometimes the "intermediate," and designated by the initials M.P.; and in quadruple engines the third cylinder is called the 2nd M.P.

Comparative Theoretical Efficiency of Various Marine Engines.—(1) *A single-cylinder expansive engine*: rate of expansion, 5; initial pressure, 80 lbs.; absolute back pressure, 4; area of piston, A.

Mean pressure . . .	= 41·76 lbs.
Effective mean pressure .	= (41·76 — 4) = 37·76 lbs.
Effective initial load on piston	= (80 — 4) A = 76 A lbs.
Efficiency of the system .	= 1·00.

(2) *A single-crank tandem compound engine:* rate of expansion, 5; initial pressure, 80 lbs. absolute; area, L.P. piston, A; ratio of cylinders, R, generally in practice, 4.

Effective mean pressure referred to L.P. piston .	= (41·76 — 4) = 37·76 lbs.
Terminal pressure in H.P., and initial pressure in L.P.	$= \frac{R}{5} \times 80 = 64$ lbs.
Effective initial load on H.P. piston	$= (80 - 64)\frac{A}{4} = 4$ A.
" " L.P. "	= (64 — 4) A = 60 A.
Efficiency of the system	= 1·00.

Total load on crank is, therefore, 64 A, against 76 A with the single-cylinder expansive engine.

As in actual practice there is invariably a drop, which will amount to as much as 10 lbs. in an engine of this kind, the initial pressure in the low-pressure cylinders being decreased by that amount, and that of the high-pressure increased. So that, actually, the total loads will be as 56·5 to 76, or a saving in the compound engine of over 25 per cent. of the load put on the rods, framing, etc.; and also enabling a large reduction to be made in the diameter of the shafting. The engine will work much more steadily, owing to the ratio of maximum to mean pressure being so largely reduced; and the handiness very much increased from the cut-off in the high-pressure cylinder being so late as $\frac{8}{10}$ the stroke. The friction of the two cylinders, etc., is, however, considerably greater than that of the one, but this is set off by the reduction in friction on the guides and journals; and the friction on the valve of the single cylinder exposed to high-pressure steam will be more than the combined friction of the two valves, the small one of which only is so exposed, while the expansion valve (which is necessary to the single cylinder for so early a cut-off) will also increase its loss from friction.

The compound engine compares more favourably with the simple expansive when both have two cylinders and two cranks. Then each engine has the same number of working parts of necessity, and the simple expansive, besides having the usual slide valves, each of which is exposed to the boiler pressure, has an expansion valve to each cylinder, in order to effect so early a cut-off. The following examples will show the results of the two systems under the same circumstances :—

(3) *A simple expansive engine having two cylinders*, each of whose pistons has an area of $\frac{A}{2}$ inches; rate of expansion, 5; initial pressure, 80 lbs.

Mean pressure	$= 41{\cdot}76$ lbs.
Effective mean pressure . .	$= 41{\cdot}76 - 4 = 37{\cdot}76$ lbs.
Effective initial load on piston .	$= (80 - 4)\frac{A}{2} = 38$ A lbs.
Effective mean load on both pistons	$= 37{\cdot}76 \times A$ lbs.
Efficiency of the system . .	$= 1{\cdot}00$.

(4) *A compound engine having two cylinders*, the ratio of whose piston area is 3, and the area of low-pressure piston, A; rate of expansion, 5; initial pressure, 80 lbs.; pressure in receiver, 21 lbs.

The cut-off in high-pressure cylinder to effect this rate of expansion $= \frac{3}{5}$ or 0·6 stroke.

The cut-off in low-pressure cylinder to maintain 21 lbs. pressure in the receiver $= \dfrac{80 \times 0{\cdot}6}{21 \times 3} = 0{\cdot}76$ the stroke.

Effective mean pressure in H.P. cylinder $= 72{\cdot}48 - 21 = 51{\cdot}48$ lbs.

„ „ L.P. „ $= 20{\cdot}32 - 4 = 16{\cdot}32$ lbs.

Effective initial load on H.P. piston . $= (80 - 21)\dfrac{A}{3} = 17{\cdot}3 \times A$ lbs.

„ „ L.P. „ . $= (21 - 4)\,A = 17{\cdot}0 \times A$ lbs.

Effective mean load on both pistons . $= 51{\cdot}48 \times \dfrac{A}{3} + 16{\cdot}32 \times A$

$= 33{\cdot}48 \times A$ lbs.

Efficiency of the system . . . $= 0{\cdot}887$.

It will be seen that the initial load on each of the pistons of the compound engine is very nearly equal in this case, and is less than half that on each piston of the expansive engine. The work done is very nearly equally divided between the two cylinders, but falls short of that done by the expansive engine by more than 11 per cent.

The compound engine, therefore, is nominally not so economic in steam, but is subject to much lighter loads, and to less variation in load and temperature.

To test the merits of the compound system carefully devised experiments were made by the British Admiralty, and by the Government of the United States, which show that, although the difference between the coal consumed per I.H.P. in the two systems was not very great, the compound engine, on the whole, is more economical. The best known of these experiments was the trial between the gunboats "Swinger" and "Goshawk," the latter having compound engines, with cylinders 28 inches and 48 inches diameter and 18-inch stroke, the former expansive engines, having two cylinders, 34 inches diameter and 22-inch stroke, and those of the sister ships given below. In these and in many other cases it was demonstrated that while on trial trips the coal and water consumptions were always less with the compound engines, it was really the results of prolonged trials on service that proved conclusively the real and practical superiority of the compound system, which was not alone in consumption of fuel per I.H.P., but in mechanical efficiency, whereby the consumption per voyage was marked, and the reduction in wear and tear even more so. These arguments were, of course, convincing to the shipowner.

Diameter of cylinders and stroke.	"Sheldrake."		"Moorhen."		"Mallard."	
	Exp., 2 cyls., 34″ x 21″.		Exp., 2 cyls., 34″ x 21″.		Comp. 31″ – 48″ x 18″.	
Boiler pressure, lbs.,	62·35	53·0	63·5	41	58·4	59·0
Vacuum, ins.,	20	24	23·3	24·4	23·75	25·1
Revolutions per minute,	115·5	84·7	121·3	92·9	124·8	98·8
Speed of piston, feet per minute,	404	295	424	324	374	296
Indicated horse-power,	367	137	387	180	398	213
Speed of ship, knots,	9·251	7·232	9·634	7·899	9·894	8·413
Water consumed per I.H.P. per hour,	21·5	25·4	20·1	24·6	17·12	17·66

Further Comparison of Efficiency of Engines.—The compound engine with three cranks and having two low-pressure cylinders and one high, possesses advantages beyond those of the two-cylinder two-crank arrangement. It was, no doubt, first chosen principally to avoid excessive diameter of low-pressure cylinders for very large power, and next because of the uniformity of the twisting moment on the shaft with the three cranks at angles of 120° apart. But, further, in this engine the work can be fairly equally divided between the cylinders without those disadvantages previously shown to exist in the two-cylinder engine working under this condition. To divide the work equally, only one-third will be allotted to the high-pressure cylinder, and one-third to each of the two low-pressure cylinders, and this can be done by maintaining a considerably higher pressure in the receiver than obtains in the two-cylinder arrangement. The "drop," therefore, is considerably less; and since each low-pressure piston has only half the area of that of the two-cylinder engine, the initial loads under these conditions are not abnormally large. The increase in receiver pressure reduces the initial load on the high-pressure piston, and hence its diameter may be increased, so as to get increased expansion in it and decrease in "drop" without increasing the initial load beyond that on the high-pressure piston of a two-cylinder engine of equal power.

(5) *An engine having two low-pressure and one high-pressure cylinders working on three cranks.*—To fully appreciate these advantages, suppose such a three-cylinder engine to be working under the same circumstances as that of example (4), page 182, so that the area of each low-pressure piston is $\frac{A}{2}$, and of the high-pressure piston $\frac{A}{3}$, the rate of expansion 5, and the initial pressure 80 lbs. absolute.

As in the previous example the cut-off is 0·6 the stroke. Suppose the pressure in the receiver to be 32 lbs. absolute,

$$\text{Then the cut-off in L.P. cylinders} = \frac{80 \times 0{\cdot}6}{32 \times 3} = 0{\cdot}56 \text{ the stroke.}$$

The effective mean pressure in the H.P. cylinder, ... $= (72{\cdot}48 - 32) = 40{\cdot}48$ lbs.

The effective mean pressure in each L.P. cylinder . . . . . $= (28{\cdot}19 - 4) = 24{\cdot}19$ lbs.

Effective initial load on the H.P. piston $= (80 - 32)\frac{A}{3} = 16 \times A$ lbs.

" " each L.P. " $= (32 - 4)\frac{A}{2} = 14 \times A$ lbs.

The effective mean load on all three pistons . . . . . $= 40{\cdot}48 \times \frac{A}{3} + 2 \times 24{\cdot}19 \times \frac{A}{2}$ $= 37{\cdot}68 \times A$ lbs.

Efficiency of the system . . . . $= 0{\cdot}998$.

It is seen by this that the initial load on the high-pressure piston is $7\frac{1}{2}$ per cent., and that on each low-pressure piston $17\frac{1}{2}$ per cent., less than the corresponding loads of the two-cylinder engine of the same size; that the gain of efficiency of the steam is $12\frac{1}{2}$ per cent., if no allowance is made for possible superheating of the steam on expanding into the receiver; the "drop" in this case is (54 — 32) or 22 lbs., as against 33 lbs. in the two-cylinder engine. It is seen, then, that theoretically this engine is nearly equal in steam efficiency, both to the simple expansive, and to the compound direct-expansion engine. On the other hand, the friction of three engines must be set against this, besides the extra cost of manufacture and the space occupied by machinery.

Experiments in the Mercantile Marine with engines having cylinders with ratios better suited to the steam pressure and other conditions demonstrated much more emphatically the superiority of the compound engine in economy of fuel and, perhaps, more decisively the other advantage of that system, for it was soon found that the wear and tear and oil consumption of expansive engines having a working pressure of 50 to 60 lbs. The two-cylinder and three-cylinder receiver type of compound engine soon became popular, and remained so until steel boilers could be made for such high pressures as to require an extension of the compound system.

Triple-Expansion Compound Engine.—The compound engine having one high-pressure, and one low-pressure, with a medium-pressure cylinder, is the one most commonly found in use, if steam of over 120 lbs. pressure is to be used economically. To ensure economy, the steam must be expanded down to about 10 lbs. absolute, the initial loads on the pistons moderate, and the "drops" not excessive. The low-pressure cylinder may be rather smaller in size than that of the ordinary *two*-cylinder compound arrangement, due to the increased efficiency of the steam from the high rate of expansion, and the greater referred mean pressure.

(6) For example:—To determine the particulars of a triple-compound engine on this system to be equal to that set out in example (4), page 182, the initial pressure being 127 lbs., and the rate of expansion 10.

The mean pressure in a single cylinder, with a cut-off at $\frac{1}{10}$ the stroke, is 41·91 lbs.; deducting from this 4 lbs. for back pressure, the mean *effective* pressure is 37·91 lbs., or nearly the same as that of example (5), page 184. Suppose the cut-off in the high-pressure cylinder is 0·6 the stroke, then the ratio of the high-pressure to low-pressure cylinder must be 6 to effect a rate of an expansion of 10. The ratio of the medium-pressure cylinder to high-pressure cylinder may be taken as $\frac{5}{2}$, and, consequently, the ratio of low-pressure to mean-pressure cylinder is $\frac{12}{5}$. The pressure between the high-pressure and mean-pressure cylinders is to be 50 lbs., the cut-off in the mean-pressure cylinder will, therefore, be 0·61 the stroke. The pressure between the low-pressure and mean-pressure cylinders is to be 21 lbs.; the cut-off in low-pressure cylinder must, therefore, be 0·6 the stroke.

The effective mean pressure in H.P. cylinder $= (115 - 50) = 65$ lbs.
,, ,, M.P. ,, $= (45{\cdot}4 - 21) = 24{\cdot}4$ lbs.
,, ,, L.P. ,, $= (19 - 4 = 15{\cdot}0$ lbs.

The effective initial load on H.P. piston . $= (127 - 50)\dfrac{A}{6} = 12{\cdot}88\,A$ lbs.
,, ,, M.P. ,, . $= (50 - 21)\,A\,\frac{5}{12} = 12{\cdot}1\,A$ lbs.
,, ,, L.P. ,, . $= (21 - 4)\,A = 17{\cdot}0\,A$ lbs.

The effective mean load on all three pistons . . . . . $\left\{ \begin{aligned} &= \left(65 \times \frac{A}{6}\right) + (24{\cdot}4 \times \tfrac{5}{12}\,A) + 15\,A \\ &= 36 \times A \text{ lbs.} \end{aligned} \right.$

Efficiency of the system . . . $= \dfrac{36}{37{\cdot}9} = 0{\cdot}949.$

It will be seen that in this case, owing to "drop," there is a loss of nearly 2 lbs., but the work is fairly divided between the three cylinders, and the initial loads are by no means high; the drop from the high-pressure cylinder is only 26 lbs., and that from the mean-pressure cylinder $9\frac{1}{2}$ lbs. This engine then effects an expansion of 10, and is, therefore, a very economic one; it will have a very regular motion, and share in all the benefits of a three-crank engine; and the stresses on the working parts will be very moderate.

(7) To see how far this is true, it is only necessary to compare these results with those from *a three-crank engine having one high- and two low-pressure cylinders*, each L.P. having a piston area of $\frac{A}{2}$.

Suppose the high-pressure cylinder to have a piston area of $\frac{A}{4}$, the cut-off in the high-pressure cylinder to effect an expansion of 10, must be 0·4 of the stroke; the receiver pressure will be 42 lbs., and the cut-off in low-pressure cylinder 0·3 of the stroke.

The effective mean pressure in the H.P. cylinder . $\Big\} = (97 - 42) = 55$ lbs.

The effective mean pressure in each L.P. cylinder . $\Big\} = (27{\cdot}7 - 4) = 23{\cdot}7$ lbs.

The effective initial load on the H.P. piston . . $\Big\} = (127 - 42)\frac{A}{4} = 21{\cdot}25 \times A$ lbs.

The effective initial load on each L.P. piston . . $\Big\} = (42 - 4)\frac{A}{2} = 19{\cdot}0 \times A$ lbs.

The effective mean load on all three pistons . . $\Big\} = \left(55 \times \frac{A}{4}\right) + 2\left(23{\cdot}7 \times \frac{A}{2}\right) = 37{\cdot}45 \times A$ lbs

The initial load on the high-pressure piston is here 65 per cent. larger than in the preceding case. and that on each low-pressure piston 56 per cent. larger than on the mean-pressure piston, and $11\frac{3}{4}$ per cent. above that on the low-pressure piston; but the efficiency of the steam is somewhat higher in the latter case, the drop from the high-pressure cylinder being only 8·8 lbs. The ratio of maximum pressure to mean in the high-pressure cylinder is 1·54, and in the low-pressure cylinder 1·6, which are about the same as those of the old expansive engine working with a boiler pressure of 45 lbs., and cutting-off at 0·3 of the stroke. It may not be always advisable to set the cranks at angles of 120° in a three-crank compound engine; their precise position should depend on the power developed in each cylinder, and the relative twisting efforts at any period.

The success of the triple-expansion engine is now so well assured, and all doubts as to its efficiency and good working are so effectually dispelled as to require no further discussion. It does not differ in any essential feature from the ordinary compound engine, and its success was in no small measure assured by the fact that most makers of the new type departed as little as possible from their previous practice in its general construction. A few years' experience demonstrated that the triple-expansion engine is more economical than the ordinary compound engine; that the wear and tear is no more, but rather less, when three cranks are employed, than with the two of the ordinary compound; and that boilers of the common marine design could be made to work satisfactorily at a pressure of 150 lbs. per square inch, and even higher, while, with ordinary care, their durability and good continued working are not less than those of similar boilers pressed to 60 lbs. per square inch under similar circumstances. Speaking generally, the consumption of fuel is 20 per cent. less with a triple-expansion engine than with an ordinary compound engine working under similar circumstances. That is, a triple-expansion engine, supplied with steam at 150 lbs. pressure, uses 20 per cent. less weight of water per I.H.P. than an ordinary compound engine supplied with steam at, say, 90 lbs. pressure, both engines being equally well-designed, manufactured, and attended to. Also, that a triple-expansion engine is more economical than an ordinary compound engine when both are supplied with steam at the same pressure, for all pressures of 95 lbs. and upwards, and especially so in the case of large engines. Hence, it may be taken that the superior economy of the triple-expansion engine is due to two causes, viz. :—(1) To the superior pressure of steam used and the higher rate of expansion thereby possible; and (2) the system whereby large initial loads and large variations of temperature in the cylinders and large "drop" in the receivers are avoided.

Increased Pressure of Steam is obtained by a very slight increase of consumption of fuel, and the efficiency of steam rapidly increases with increased pressure; hence, steam of high pressure is more economical than that at inferior pressures. For example :—

(i.) The total heat of evaporation of 1 lb. from 100° and at 274° F. (corresponding to 45 lbs. pressure absolute) is 1,097 thermal units.

(ii.) From 100° and at 320° F. (corresponding to 90 lbs. absolute) is 1,110 thermal units.

(iii.) From 100° and at 353° F. (corresponding to 140 lbs. absolute) is 1,120 thermal units.

(iv.) From 100° and at 377° F. (corresponding to 190 lbs. absolute) is 1,127 thermal units.

Suppose in each case the steam to be expanded to a terminal pressure of 10 lbs. absolute, the rates of expansion will then be 4·5, 9, 14, and 19 respectively; and the mean pressures corresponding to these initial pressures and rates of expansion will be 25 lbs., 32 lbs., 36 lbs., and 39 lbs. respectively. If the volume of a pound of steam varied exactly in the inverse ratio of the pressure, these figures would represent the relative values of the efficiency of the steam at the various pressures. But, taken exactly, the relative values are 25, 33·3, 38·5, and 42·6, thus showing that a pound of steam at 90 lbs. pressure is capable of doing 33 per cent. more work than a pound at 45 lbs.; a pound of steam at 140 lbs. pressure, 16 per cent. more than a pound at 90 lbs.; and a pound at 190 lbs. pressure, 10·6 per cent. more than a pound at 140 lbs. pressure, or 28 per cent. more than at 90 lbs. pressure. In other words, an engine using steam at 140 lbs. pressure should, apart from any practical considerations, consume 16 per cent. less fuel than one using steam at 90 lbs.; and, again, that an engine using steam at 190 lbs. should consume 28 per cent. less fuel than one using steam at 90 lbs., and 10·6 per cent. less fuel than one using steam at 140 lbs.

Looking to see how far practice agrees with these results, it is found that the ordinary compound engine, using steam at 140 lbs., is only a little more economical than one using steam at 90 lbs., while the triple-expansion engine, with steam at 140 lbs. pressure, gives an economy rather greater than theory shows should be due to increased pressure. It follows, then, that there is some other cause operating to produce the economic results shown in every-day practice with this system, for there is now no question that the saving in fuel effected by a triple-expansion engine, using steam and expanding 11 or 12 times, is about 20 per cent. of that used by an ordinary compound engine of the same power, using steam at 90 lbs. and expanding 8 to 9 times.

It used to be maintained by the opponents of triple-expansion engines that, if the ordinary compound engine is designed with a long stroke, it is as economical. In order to see how far this is true by practice, it is sufficient to examine the diagrams of the engines of the s.s. "Northern," whose cylinders were 26 inches and 56 inches diameter and 60-inch stroke, used steam at 130 lbs. absolute, and indicated 1,235 H.P., which show a consumption of 15·4 lbs. of water per I.H.P. per hour; and those of s.s. "Draco," whose engines had cylinders 21 inches, 32 inches, and 56 inches diameter and 36-inch stroke, using steam at 125 lbs. absolute, and indicated 618 H.P., and showed a consumption of 14·1 lbs. per I.H.P. per hour. Or, again, by comparing the performance of the "Draco" with that of the "Kovno," whose cylinders were 25 inches and 50 inches diameter and 45-inch stroke, using steam at 105 lbs. pressure absolute, and indicating 809 H.P., with a consumption of 16·6 lbs. of water per I.H.P. In these cases, the consumption of the "Kovno" was 17·73 per cent. in excess of that of the "Draco," and 7·58 per cent. in excess of that of the "Northern"; and the "Northern" consumed 9·4 per cent. more than the "Draco."

It is, however, needless now to multiply cases, as it is a matter of common observation that the saving in fuel is from 20 to 25 per cent., and it may safely be taken that the triple-expansion engine, using steam at 165 lbs. absolute pressure, uses 20 per cent. less fuel than an equally good ordinary compound engine of equal power and working under similar circumstances, using steam at 100 lbs. absolute pressure.

As the three-crank triple-expansion engine is now accepted as the most suitable for marine practice generally, it is instructive to compare it, so far as practical considerations are concerned, with the ordinary compound. For that purpose, suppose, now, two engines are to be taken—viz., a triple-expansion and an ordinary compound—to develop equal powers with the same stroke of piston and same diameter of low-pressure cylinder. The initial pressure in the one case is to be 100 lbs. absolute, and in the other 165 lbs. absolute. Let the number 14 represent the area of the low-pressure piston in each case; the mean pressure referred to the low-pressure piston is to be 24, and the efficiency of the systems equal, so far as the steam is concerned. The area of the high-pressure piston may then, without fear of controversy, be taken as 4 for the ordinary compound, and 2 for the triple, and the area of the medium-pressure piston of the triple as 5. If the referred mean pressure is equally divided in each case between the cylinders, the mean pressure in H.P. of the compound engine will be $\frac{14}{4} \times 12$, or 42 lbs.; in the triple-expansion engine the mean pressure in the H.P. cylinder will be $\frac{14}{2} \times 8$, or 56 lbs.; and in the M.P. cylinder $\frac{14}{5} \times 8$, or 22·4 lbs., and the following shows the relative work done, viz. :—

Ordinary Compound Engine.		
High-pressure cylinder,		4 × 42 or 168
Low	,,	14 × 12 or 168

Triple-Expansion Engine.		
High-pressure cylinder,		2 × 56 or 112
Medium	,,	5 × 22·4 or 112
Low	,,	14 × 8 or 112

That is, the *average* load on the rods, columns, guides, etc., is 50 per cent. more with the ordinary compound engine than with triple-expansion.

To obtain a mean pressure of 24 lbs., with an initial pressure of 165 lbs. absolute, and a pressure in the condenser of 2 lbs., the rate of expansion is 14, with an efficiency of 0·6; and with an initial pressure of 100 lbs. the rate of expansion is 7.

On examining diagrams taken under these circumstances from actual engines, the following is to be observed :—

Initial pressure in the high-pressure cylinder of the compound engine, 98 lbs.; back pressure, 23 lbs.; effective initial pressure, 75 lbs. per square inch; or load on the piston, 75 × 4, or 300. In the low pressure the initial pressure is 22 lbs.; back pressure, 4; giving an effective initial pressure of 18 lbs. per square inch; or load on the piston, 18 × 14, or 252.

The initial pressure on the high-pressure cylinder of the triple-expansion engine is 160 lbs.; back pressure, 63 lbs.; effective initial pressure, 97 lbs.; or the load on the piston, 97 × 2, or 194. In the medium pressure the initial pressure is 70 lbs.; and the back pressure, 21 lbs.; effective initial pressure, 49 lbs.; or the load on the piston, 49 × 5, or 245. In the low pressure the initial pressure is 18 lbs.; and the back pressure 4 lbs.; giving an effective initial pressure of 14 lbs. per square inch; or a load on the piston of 14 × 14, or 196. Thus showing the loads in the case of the triple-expansion engine to be much less, notwithstanding the higher pressure of steam employed.

This, too, is shown in actual practice by comparing the initial loads on the engine of the s.s. "Northern," whose cylinders are 26 inches and 56 inches diameter and 60-inch stroke, indicating 1,242 H.P., and supplied with steam at 130 lbs. pressure absolute, with those of the triple-expansion engine of the s.s. "Ariel," whose cylinders are 23 inches, 35 inches, and 60 inches diameter and 57-inch stroke, indicating 1,527 H.P., and supplied with steam at 165 lbs. pressure absolute.

The "Northern's" high-pressure piston sustains an initial load of 530 × 100, or 53,000 lbs.; the low-pressure, 2,463 × 24, or 59,112 lbs. The "Ariel's" high-pressure piston sustains 415 × 100, or 41,500 lbs.; the medium-pressure, 962 × 60, or 57,720 lbs.; and the low-pressure, 2,827 × 18, or 50,886 lbs.—notwithstanding that the engines are larger and develop nearly 25 per cent. more power with higher boiler pressure.

The more even distribution of pressure also very materially affects the resistance of the slide-valves, and so tends in every way to reduce the losses due to mechanical causes.

Experience has shown that the wear and tear of the triple-expansion engine with three cranks is very considerably less than that of ordinary two-crank compound engine of the same power and stroke, and no doubt this is due to those causes already shown to exist with this class of engine, as compared with the expansive engine.

The three-crank triple-expansion engine has, however, shown another valuable quality, and one which may easily be surmised from the foregoing reasoning—viz., that a much higher indicated power may be developed with a low-pressure cylinder, equal in size to that of a common compound engine, without any increase in the initial loads on the pistons. This may be shown by comparing the performance of the engines of the "Eldorado," whose cylinders were 26 inches, 40 inches, and 68 inches diameter and 39-inch stroke, supplied with steam at 165 lbs. absolute pressure, with that of the engines of the "Juno," having cylinders 35 inches and 69 inches diameter and 39-inch stroke, supplied with steam at 100 lbs. absolute pressure. When running at 72 revolutions, the "Eldorado's" engines develop 1,572 I.H.P., and the "Juno's" 1,249 I.H.P., or 26 per cent. more power from the triple-expansion than from the ordinary compound, although the low-pressure cylinder was 3 per cent. smaller. The initial loads on the pistons of the "Eldorado" are 54,060 lbs., 66,568 lbs., and 61,727 lbs., as against 69,264 lbs. and 71,041 lbs. on those of the "Juno."

Similar results can be shown with four-crank engines of various sizes; and to extend the question, it may be taken as approximately correct that a referred mean pressure may be used in a triple-expansion engine 50 per cent. higher than with an ordinary compound engine without any serious difference in the stresses on the working parts and frame-work. It is for this reason possible to manufacture a triple-expansion engine at the same price per I.H.P. as an ordinary compound engine. The propelling efficiency of the three and four-crank engine is especially noticeable when running at low speeds, and it is no doubt on this account they were best for naval purposes where so much cruising is done at comparatively slow speeds. They are also capable of being worked at much fewer revolutions without stopping on the centres than a two-crank engine, which is highly advantageous in navigating intricate channels and docks and during a fog, as steerage power is kept without much "way" on the ship. They are also, when well constructed and properly adjusted, almost noiseless, and cause little or no vibration, which is an advantage in every ship, but especially in yachts and passenger steamers.

The Compound Systems of Cylinders are now admitted on all hands without any further controversy to be superior to the simple-expansive one, notwithstanding that in the past at each of its stages of development this claim has been contested by those who seemed unable to grasp the fact that the determining factors in each particular controversy were practical rather than theoretical ones, and much of the same nature as the one which caused Watt to remove the condensation of the steam from the cylinder to the condenser. Moreover, it was by practical tests rather than argument in each of these stages that engineers were weaned from their love of the old to their faith in the new system. It is unnecessary now to dwell on the subject, or to recapitulate further the early experiments made to prove that the compound engine was superior to the simple expansive, as measured by the water consumption per I.H.P.; or those later ones, whereby the compound was shown to be inferior to the triple-expansion engine; nor even to those later still, when the advocates of a further subdivision claimed for their quadruple-expansion engine a distinct improvement on the good results achieved by the triple.

So far no one has ventured on a quintuple stage system, although engines with five cylinders and cranks have been made, besides which the increase in boiler pressure now possible with cylindrical boilers has warranted such an advance. Although the controversy has ceased, and a compound system of cylinders of some kind is always found in a modern marine reciprocating steam engine, it is well that the reasons for such an adoption should be clearly stated and well understood, for after all it is somewhat in the nature of promulgating a paradox to proclaim that if steam is expanded through all the ports, passages and pipes of a compound system of four stages, the economic result is better than if expanded in a single cylinder in the simplest way possible.

The fact is that, theoretically, the compound system is wrong so far as efficiency of steam expansion is concerned; nevertheless, in practice, the same conditions which caused the compound engine to triumph over the expansive have established the superiority of the triple over the compound, and given an advantage to the quadruple over the triple. The system with its extensions have permitted the safe employment of higher steam pressures with their higher efficiencies without making increases in the initial loads on the rods, crank-pins, bearings, guides, etc., and, more important still, without the large variations in temperature in the cylinders so objectionable when steam expansion is to be effected without loss.

When the boiler pressure was 30 lbs. (45 lbs. absolute) the steam at entry to the cylinder would have a temperature of about 270° F.; during exhaust the temperature in the cylinder would be only about 150° F., or a difference during the cycle of 120°. The expansive engine using steam at 75 lbs. absolute would have a range throughout the system of 155°, while in each cylinder of the compound engine it will be only about 70°, allowing for the usual drop of pressure between them. In the triple-expansion system using steam of 195 lbs. absolute pressure, although the variation throughout is about 230°, in each cylinder it is not much more than 70°.

In the case of a compound engine, the L.P. cylinder is about the same size as the single cylinder of an expansive engine; or, supposing the latter to be a two-crank engine, each of its cylinders will be half the capacity of the low pressure of the compound, and so each piston is half the area of that of the L.P. piston. The initial pressure per square inch on the pistons of the two engines will be the same if the boiler pressures are equal (say 60 lbs.); but the area of the high-pressure piston of the compound would be only about 0·36 of the low-pressure, and consequently the ratio of forward load on it would be under these conditions as 36 to 50 of that on each of the expansive engine cylinders, but the *back* pressure of these latter would be only about 3 lbs., so that their effective initial load would be $(60 + 15 - 3)$, or 72 lbs. per square inch, while the high-pressure piston of the compound would have a back pressure of about 22 lbs. absolute; consequently the effective maximum load would be in this case $(60 + 15 - 22)$, or 53 lbs. per square inch only. The total loads will be then measured relatively as—

Compound, H.P. piston, . .	$36 \times 53 = 1908$
Expansive, each piston, . . .	$50 \times 72 = 3600$

The L.P. cylinder of the compound engine will have a forward initial pressure on it of about 18 lbs. absolute, and a back pressure of 3 lbs., the effective initial pressure being thus 15 lbs. Then the following holds good for comparison as being the maximum loads :—

On each of the cylinders of the expansive engine the load may be taken as	$50 \times 72 = 3600$
That on the H.P. cylinder of the compound engine,	$36 \times 53 = 1908$
That on the L.P. cylinder of the compound engine,	$100 \times 15 = 1500$

But not only is the range of temperature greater with the cylinders of the simple engine, but all the steam that enters must pass over a surface, which, for 45 per cent. of the cycle time, is exposed to the cooling action of the condenser; moreover, those ports, passages, valves, cylinder ends, and pistons have a surface three times the area of the corresponding parts of the H.P. cylinder of the compound system, over which all its steam passes, and, moreover, its exposure is only to that of the receiver, whose temperature will be about 230°, as against 140° of the condenser. The difference will be greater still with a modern condensing apparatus, providing a vacuum as high as 29 inches, with a temperature during exhaust probably as low as 110°. Under such conditions the condensation on entry to the cylinder, and during expansion in it, would be great; and, besides, beyond the loss of heat, the efficiency of the engine would be lowered by the impeding action of the deposited water. Any re-evaporation that is possible would take place too late in the stroke to be of any practical use in the cylinder of the simple engine; but in the case of the compound, however, it would be of considerable use in the second cylinder, although no value may have accrued from it in the H.P. cylinder where the transformation took place. The loss by condensation at the valves, whether they be of the flat or piston type, is more considerable in practice than might have been anticipated. The inside or exhaust side of the L.P. valve of a compound, as of both of those of the simple engine, is exposed always to the cooling effect of the condenser, while the hot steam is on the other side; under these circumstances, and inasmuch as the metal of these valves is comparatively thin, the condensation is really quite considerable; but it is even more so in the simple engine, as the steam is then of boiler pressure or nearly so, and consequently of much higher temperature, so that transmission is the more rapid due to the greater difference of temperature.

In fact, the difference of temperature is the all-ruling factor in this as in some other engineering problems; for, beside the above, on the practical side there is always the additional risk of fracture of cylinder and valves, and, in the case of a compound system, the straining action on it of the cylinders, if they are cast or rigidly bolted together. The solution of the latter difficulty, however, is easy, being simply to avoid rigid connections, and provide that each cylinder shall be quite free to expand or contract as circumstances may cause it.

Fig. 65 is interesting and instructive, showing, as it does, the economic process of expanding steam of high pressure through the four cylinders of a quadruple-expansion engine made by Messrs. Richardson & Westgarth. The stages can be clearly traced on the combined diagram, and the temperature and pressure changes marked.

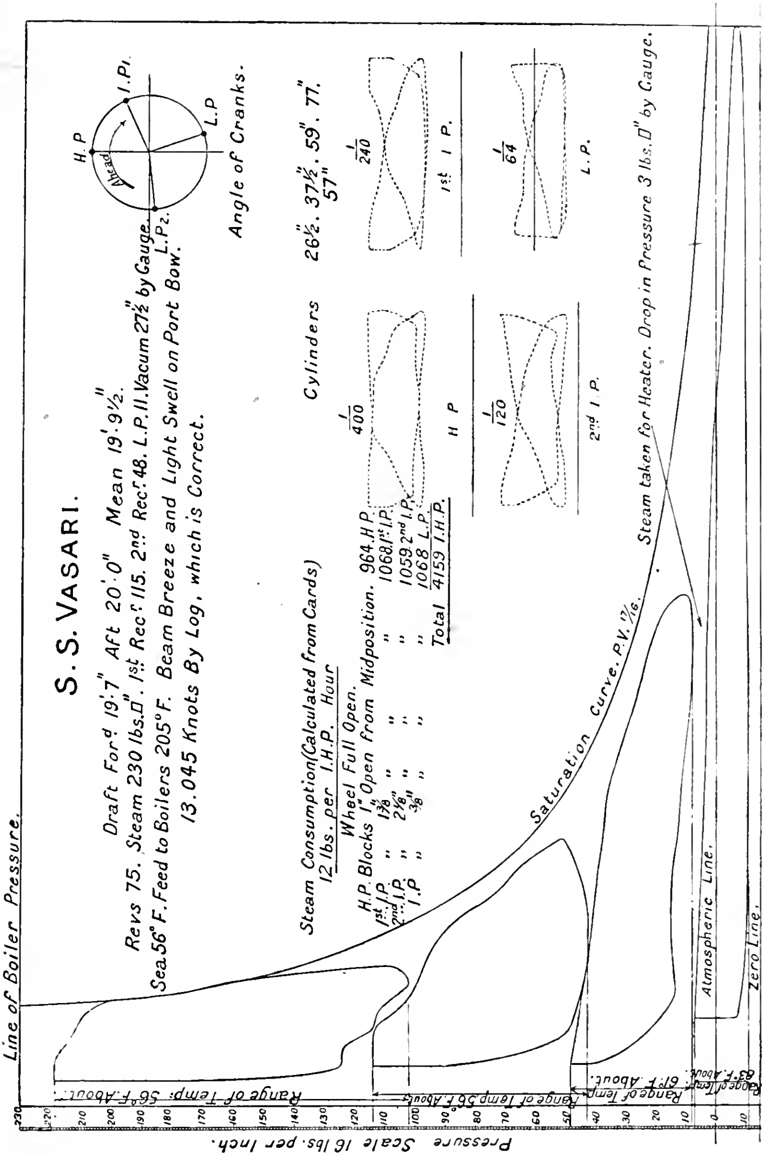


Fig. 65.—Indicator Diagrams and their Combination of the Quadruple-expansion Engines by Richardson, Westgarth & Co., Limited.

Fig. 66 is also instructive, as by it can be seen the various stages of progress in the use of steam on shipboard, and the effect of adding a L.P.

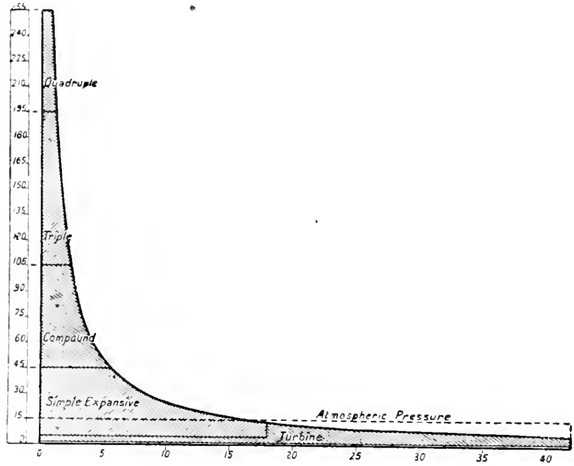


Fig. 66.—The various stages in the use of Steam expansively in Marine Engines.

turbine to a compound cylinder engine studied, and how the attenuated and apparently feeble steam at exhaust is capable of producing in a suitable generator quite a large amount of power.

CHAPTER VIII.

HORSE-POWER—NOMINAL, INDICATED, AND SHAFT OR BRAKE.

WHEN the steam engine began to replace other motors, it was soon found necessary to introduce some unit by which its power could be expressed without using such high numbers as foot-pounds ran to, as to place it beyond the grasp of ordinary minds. As the engine was frequently taking the place of horses to operate mining and other machinery, it was only natural that the work performed by a horse should be taken as the basis for this unit of measurement. The number of units of work performed by the most powerful dray horses in a minute was found to be 33,000, so Watt chose this as the unit of power for his engines, and called it "horse-power," and this has continued to be the standard ever since, both for land and marine engines.

Watt found that the mean pressure usually obtained in the cylinders of his engines was 7 lbs. per square inch. He had also fixed the proper piston speed at $128 \times \sqrt[3]{\text{stroke}}$ per minute, and his engines were arranged to work at this speed, so that he estimated the power which would be developed when at work to be

$$\text{Area of piston} \times 7 \times 128 \sqrt[3]{\text{stroke}} \div 33{,}000.$$

The power so calculated was called "*Nominal*," because the engine was denominated as of that power, and in practice that power was actually obtained. But when the boiler could be constructed so as to supply steam above the atmospheric pressure, and the engine ran with more strokes per minute than before, the power actually developed exceeded the nominal power, and from the name of the instrument by which the pressure of steam in the cylinder was obtained it came to be called the "*Indicated*" Power.

The discrepancy between Nominal and Indicated Power became in time so great, that for all scientific purposes the former ceased to be of value. It remains, nevertheless, in general use among manufacturers and users of engines, because it better conveys the commercial value and size than does the developed power; for since the area of the piston is usually the only *variable* in the expression, it follows that the size of the cylinder, and therefore the size of all the other parts, must vary directly with the Nominal Horse-Power. But since Indicated Horse-Power depends on three functions—viz., area of piston, speed of piston, and the pressure of steam—the value may be changed by altering the value of one or more of these, which alteration may be material without affecting the commercial value. For example, an engine may be caused to run at a much higher number of revolutions, even so as to double its Indicated Power, with hardly any additional cost whatever in construction to the engine itself.

The Admiralty modified Watt's rule to suit the practice of the early days of steam-navigation, by substituting the actual piston speed for the arbitrary one, and so the old rule was for

$$\left.\begin{array}{c}\text{Admiralty Nominal}\\ \text{Horse-Power}\end{array}\right\} = \frac{\text{Area of piston} \times \text{speed of piston} \times 7}{33{,}000}.$$

The Admiralty, however, dropped the use of the expression altogether, but, before doing so finally, used it in the modified sense of being one-sixth of the Indicated Power.

In the Mercantile Marine the rule for Nominal Horse-Power is by no means uniform. Before the introduction of the compound engine and increased boiler pressures, it was usual to allow 30 *circular* inches per N.H.P. —*i.e.*, the rule was—

$$\text{N.H.P.} = \frac{(\text{Diameter of cylinder in inches})^2}{30} \times \text{number of cylinders.}$$

For two-stage compound engines—D being the diameter of the low-pressure cylinder, and d that of the high-pressure—

$$\text{N.H.P.} = \frac{d^2 + D^2}{33}.$$

But neither of these rules took into account the length of the stroke or the boiler pressure, although it was generally understood and roughly standardised. The general adoption of the triple-compound engine, however, caused the question of stroke to be removed from the field of competition, and there is now more uniformity in practice. Besides which, in some large centres of marine engineering, all the makers have a standard set of sizes of cylinders for N.H.P. to which they adhere. In order to meet difficulties some engine-makers have adopted a rule for Nominal Horse-Power, based on the *capacity* of the cylinder, and in so doing have nearly met the requirement on which the continuance of the expression depends—viz.. that it is a measure of the commercial value of the engine. The power *per revolution* depends on the capacity of the cylinder so long as the mean pressure is the same; but since small engines are usually worked at a higher number of revolutions than larger ones, the power developed by the former will bear a larger ratio to the Nominal Power than will be the case in the latter.

A simple and fair rule for N.H.P.* is deduced as follows:—D is the diameter of low-pressure cylinder, d that of the H.P., d_1 that of the M.P., and d_2 that of the second intermediate of a quadruple engine; S is the stroke, all in inches, and $D \times 0{\cdot}65$ is the standard length of stroke; in a triple compound $\left(\frac{D}{d_1}\right)^2 = 2{\cdot}7$; $\left(\frac{D}{d_2}\right)^2 = 6$ or thereabouts; in a quadruple compound $\left(\frac{D}{d}\right)^2 = 8$; $\left(\frac{D}{d_1}\right)^2 = 4$; $\left(\frac{D}{d_2}\right)^2 = 2$ or thereabouts.

$$\text{N.H.P. (Triple)} = \frac{d^2 + d_1^2 + D^2}{30}.$$

$$\text{N.H.P. (Quadruple)} = \frac{d^2 + d_1^2 + d_2^2 + D^2}{30}.$$

* The Board of Trade Rule now in force is N.H.P. $= (3\,H + D^2 \sqrt{S}) \times \sqrt[3]{P} \div 700$.

H is the total heating surface, D the diameter of L.P. cylinder, S the stroke in feet, and P the load on safety valve in lbs. per square inch.

Any change of stroke should give a proportionate change of N.H.P., hence

$$\text{N.H.P.} = \frac{d^2 + d_1^2 + D^2}{30} \times \frac{S}{D \times 0{\cdot}65}.$$

Substituting the values of d, d_1 and d_2 in terms of D, then

$$\text{N.H.P.} = \frac{D \times S}{15} \text{ for a compound engine.}$$

$$\text{N.H.P.} = \frac{D \times S}{12{\cdot}6} \text{ for a triple-compound engine.}$$

$$\text{N.H.P.} = \frac{D \times S}{10{\cdot}5} \text{ for a quadruple-compound engine.}$$

Lloyd's N.H.P.—No *Nominal* Power, however, can be any guide to the capabilities of the engine, unless the power of the boilers is also in some way expressed or understood; and as it is not easy to imagine how the former can be introduced into any expression which shall effect the latter, or *vice versâ*, the suggestions of Lloyd's Committee remained unfulfilled, but the Register now contains a statement of the leading particulars of the boilers, and for purposes of levying the fees for the Survey and Registration of Machinery, Lloyd's Register employ the following rule,

$$\text{Lloyd's nominal horse-power} = p \times z \left(\frac{D^2 \sqrt{S}}{100} + \frac{H}{x} \right),$$

where p is the boiler pressure; D is the diameter and S the stroke of L.P. piston, both in inches; and H is the heating surface in square feet.

The value of x is 15 for the ordinary boiler with natural draught, but with forced or induced draught x is 12.

The value of z is 0·34 when the boiler pressure is under 160 lbs.; when it is 160 lbs. per square inch or above z is 0·393.

That there is need of uniform practice in naming the power of engines, is apparent to everyone having to do with steamships, and the Board of Trade Department, which registers the power, has so far limited its efforts in this direction to the old formula, N.H.P. $= \dfrac{d_1^2 + d_2^2 + d_3^2}{30}$. As this takes no cognisance of stroke, it was never satisfactory; the Department, however, is still satisfied with it for its purpose, but it is surely time to find some other rule which shall determine the rating of engineers and such other matters, as well as be a fair indication of the power the ship possesses to propel her.

Estimated Horse-Power.*—As it is desirable that a power be named for an engine which shall enable the lay mind to judge of its capabilities, probably the better plan would be to revert to the principle of Watt, who, as has been shown, attempted to specify the power which the engine was actually expected to develop; such a rule, therefore, should give approximately the Indicated Horse-Power. It would, of course, be far better to register the I.H.P., but as it is not always possible to obtain this, the next best method is to estimate it, and call it the Estimated Horse-Power, or E.H.P.

* N.E. Coast I. E. and S. recommend the following :—

$$\text{E.H.P.} = \frac{D^2 (S \quad 4)}{21{\cdot}9}. \qquad \text{D in inches, S in feet.}$$

The following rule will give approximately the horse-power developed at full speed by a two-stage, triple-, or quadruple-expansion engine made in accordance with modern practice :—

$$\text{E.H.P.} = \frac{D^2 \times \sqrt{p} \times R \times S.}{7{,}800}.$$

Where D is the diameter of the low-pressure cylinder, p the absolute pressure, R the number of revolutions per minute, S the stroke of piston in feet.

For Example.—To estimate the Indicated Horse-Power of an engine having cylinders 30 ins., 48 ins., and 80 ins. diameter and 48-ins. stroke, revolutions 75, and boiler pressure 165 lbs.

$$\text{E.H.P.} = \frac{80^2 \times \sqrt{180} \times 75 \times 4}{7{,}800}$$
$$= 3297.$$

Many other rules have been propounded for N.H.P., some of which are ingenious, but impracticable, while others fail to give results of any value whatever, so that neither class needs notice here ; but it may be mentioned that when *non-condensing* engines were more used in steamships than they are at present it was found necessary to have a special rule for them, which was

$$\text{N.H.P.} = \frac{D^2 \times \sqrt[3]{S}}{20}.$$

D being the diameter of the cylinder in inches, and S the stroke in feet.

Indicated Horse-Power may be defined as the measure of work done in the cylinder of a steam-engine, as shown from the indicator-diagrams, and only falls short of the actual work by such small losses as are caused by the friction of the pin or pencil against the paper, the friction of its working parts, and that in the pipes or passages connecting the indicator to the cylinder. The latter discrepancy is by far the most important, and is sometimes serious in very long stroke engines, where the indicator pipe is several feet long. The others, in the hands of a skilful operator, are not so serious, certainly not in modern marine engines to the extent stated by Mr. Hirn, who says he found the Indicated Horse-Power, owing to losses in the diagram from the friction of the indicator, to correspond with the *useful work* done by the engine. All the same, it should not be forgotten that with such an instrument as the indicator, the nearer it is to the steam in the cylinder the better. There should be no pipes, if possible, and, if any, they should be fairly large.

The Indicator-Diagram.—The diagram itself shows only the pressure of steam acting on the piston at any and every part of its stroke ; but from it may be calculated the mean effective pressure acting during that stroke, and it is assumed that the particular diagram measured is only a sample of what might have been taken at every stroke, so that the mean pressure thus calculated is the force acting on the piston during the whole period of its motion in which the power is taken—usually one minute. Hence, Indicated Horse-Power = area of piston in inches $\times$ mean pressure in lbs. per square inch $\times$ number feet travelled through by the piston per minute $\div$ 33,000.

This, of course, applies only to double-acting engines, as in single-acting engines the pressure is acting only half the time on the piston, and hence, instead of taking the number of feet travelled through by the piston per minute as the multiplier,—*the length of stroke in feet* × *number of strokes per minute* should be substituted.

Mean Pressure.—The mean pressure is usually obtained by dividing the indicator-diagram by a number of equidistant ordinates perpendicular to the atmospheric line, and so placed that the distance of the first and last from the extreme limits of the diagram is half the distance between two consecutive ones; the sum of their lengths, intercepted by the diagram, divided by their number, gives the mean length, and this, referred to the scale on which the diagram was drawn, will give the mean pressure. To illustrate this:—Fig. 67 is an indicator-diagram whose length, A X, is, say, 5 inches, and taken with a spring requiring a pressure of 30 lbs. per square inch to compress it 1 inch; so that if M L is 2 inches, it represents a pressure of 60 lbs.; and if B L is $2\frac{1}{2}$ inches, it signifies that, at the point L, the pressure on the piston was $2\frac{1}{2} \times 30$ lbs., or 75 lbs. per square inch above the line A X, which, in this case, shall be *the line of no pressure*, and hence is 75 lbs. *absolute*, or 60 lbs. above the atmospheric pressure. Now, for convenience of division, let there be 10 ordinates enclosing 9 spaces—since there is to

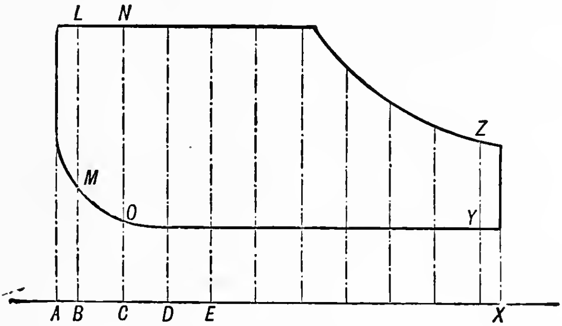


Fig. 67.—Indicator Diagram.

be a half space at each end, there will be in all equal to 10 spaces—so that the distance between the ordinates is 5 inches ÷ 10, or half an inch. Measure off A B = $\frac{1}{4}$ inch; B C, C D, D E, etc., each = $\frac{1}{2}$ inch, and at B, C, D, E, etc., draw perpendicular lines, cutting the diagram at M L, O N, etc., Y Z. Then (M L + O N + etc. . . . . + Y Z) ÷ 10 = x inches, and $x \times 30$ is the mean pressure of the diagram.

This diagram is from one side of the piston only, and, when one only is obtainable, it is sometimes assumed to represent both, and the mean pressure thus obtained used to calculate the power; but it seldom happens, although it is much to be desired, that the mean pressure is *precisely* the same on both sides of the piston, consequently, any result obtained in this way is not satisfactory. If the effective area of the piston is the same on both its sides —that is, if there is the same area on which the steam acts to propel the piston forward, on the one side as on the other—the mean pressure found from the diagram taken from the one side, may be added to that found from the diagram taken from the other, and divided by 2 to give the true mean pressure per revolution.

Professor Rankine showed that the diagram should be divided from A to N by ordinates equidistant apart, and the mean obtained by the following rule:—Let n be the number of such divisions (usually 10), b_0, b_1 b_2 b_3 . . . b_n the length of the ordinates intercepted by the diagram: then

$$\text{Mean length} = \left(\frac{b_0 + b_n}{2} + b_1 + b_2 + \text{etc.}, \ldots + b_{n-1}\right) \div n.$$

The chief objection to this is that, in actual practice b_n would be always without value, and b_0 either without value, or so difficult to measure as to cause differences of opinion as to its value.

Another, and a very ready way of obtaining the mean pressure, is by measuring the *area* of the diagram by a *planimeter*, and dividing it by the length A N, the result being the mean breadth as before, and this multiplied by the scale of lbs. of the spring will give the mean pressure. This is, of course, the quickest plan, and the most accurate, as being mechanical; and where many diagrams have to be calculated with despatch, it is very advisable to have a good planimeter. Special instruments are now made for this purpose.

In whatever way the mean pressure be measured, it forms the basis of calculation of actual energy, or, as it has been called, Indicated Horse-Power, and is therefore of the utmost importance, since most modern formulæ bearing on marine machinery and marine propulsion are based on I.H.P. Hence, any error in taking the diagrams must lead to errors in design from calculations by formulæ based on false premises; this should always be borne in mind by the operator, on whose skill and care a good and true diagram depends as much as on a good instrument. It would be a very valuable quality in an indicator to be able to give the *useful work* of the engine, as was stated by Mr. Hirn to be the case generally; but it is a quality which no such instrument can possess, inasmuch as, with the same *cylinder performance*, there may be a great variety *of actual performances* of the engine, depending on the efficiency of the various parts, and the indicator only gives this cylinder performance. Could the effective power be easily obtained as it is with electric generating engines, a great benefit would be conferred on marine engineers in making calculations, etc., and in determining the best make and design of engine. The *precise* power absorbed in overcoming the resistance of the working parts of a marine engine could not be measured; for if the engine be allowed to run without load, it is not running in the *same* state as when running with its load; and any diagrams taken then cannot be taken as the loss from friction, etc., on the guides, journals, pistons, and valves when running with the increased pressure on them due to the increased pressure on the piston with the load. Since the efficiency of an engine very much depends on the resistance on these parts, any calculation or formula which excludes, or does not give due allowance to this, is misleading. Hence, to assume that the power absorbed by an engine in overcoming its resistance is measured by the power indicated when running without load is not correct—as it is possible for an inefficient engine to show a high efficiency when tested in this manner. It is, however, true that a large portion of the resistance is the same, or practically the same, when the engine is running at the same number of revolutions with and without load. The frictional resistance of glands, pistons, pumps, tunnel-shaft journals, high-pressure piston valve, and sundry small gear, is the same *per revolution* at any speed, whatever be the load, or power developed. The total resistance or non-useful work of engines, therefore, probably varies nearly directly as the revolutions.

Shaft Horse-Power is a term now often used and well known to the marine engineer, and is likely to be more so than the I.H.P., which hitherto has been in such general use to express the performance of an engine.

Inasmuch as the indicator was of no service to the maker of turbines, nor could any instrument which merely shows the pressure of steam be a means of determining the power developed by a velocity machine, some other method had to be devised for that purpose. When a turbine was driving a dynamo it was easy to calculate the mechanical power of the driver by the measure of the electrical output from a dynamo whose efficiency was known. The brake horse-power of a turbine could be found by causing it to operate on a water brake, and so for some considerable time that was the only way in which turbine power was determined. It is true when there were two sister ships, whose actual resistance was known, an approximation to the power developed by the turbine was made by means of that of the reciprocating engine in the other ship, as given by the indicator. This, however, was not a satisfactory state of things, especially as I.H.P. is looked on with suspicion, and that not without justification, for even with a slow-running engine the human element is a factor involved in the accuracy of the I.H.P., while with a very fast-running engine the diagram made by the best of indicators handled by a man is influenced by the skill of the operator. There is, however, an indicator, the product of Prof. Hopkinson's genius, that does give a diagram which is free from suspicion, by which the actual cycle of pressure in a cylinder may be viewed as it really is, and from it the power may be determined. But it is as inapplicable to the turbine as the Richards or other indicator for the reasons given above.

It was necessity, therefore, which stimulated invention and caused to be brought forth the torsion meter, whereby the power transmitted by a shaft can be determined by observing the angle of twist of a definite length of that shaft. There are various kinds of such meters, all of which show a considerable amount of inventive genius, mechanical knowledge, and skill. In some of them measurement is mechanical, made by a self-registering instrument, in others the eye is employed, and in others the ear, to fix the amount of distortion at any given time when the shaft is transmitting energy. It is certainly most desirable to eliminate the human element if possible, but in doing so at present other factors of an equally undesirable kind are introduced (*v.* Chap. v.).

Shaft Horse-Power calculated as on p. **150** :—d is the diameter of a shaft, l the length under observation, both in inches, T is the torque in inch-pounds, θ is the angle of twist in the length l, and Mr is the modulus of rigidity, which is about 11,750,000 in solid shafts and 12,150,000 in hollow ones.

$$\theta \text{ in degrees} = \frac{584 \times T \times l}{Mr \times d^4} \text{ for solid shafts}$$

$$= \frac{584 \times T \times l}{Mr\,(d^4 - d_1^4)} \text{ for shafts with a bore of diameter } d_1.$$

The Shaft Horse-Power at revolutions $R = \frac{T}{12} \times \frac{2 \pi R}{33,000} = T \times R \div 63,000.$

The formula in daily use is therefore (*v.* p. 150)

$$\text{S.H.P.} = \frac{\theta d^4 \times R}{Q \times l} \text{ for solid and } \theta (d^4 - d_1{}^4) \frac{R}{Q \times l} \text{ for hollow shafts.}$$

Q is usually taken at 3·27; this assumes the modulus to be 11,250,000, which is below what is generally found for steel shafts of very good quality.

Example 1.—What is the total H.P. of a ship having three shafts 6 inches in dameter revolving at 600 times per minute, and twisting 0·4° in 40 inches?

$$\text{S.H.P. of each shaft} = \frac{0{\cdot}4 \times 1{,}296 \times 600}{3{\cdot}27 \times 40} = 2{,}378.$$

The total power of the ship is, therefore, 3 × 2,378, or 7,134 H.P.

Example 2.—A tunnel shaft is revolving 300 times per minute, and is 10 inches diameter; the angle of twist in 40 inches is 0·33°, what power is it transmitting?

Here $$\text{S.H.P.} = \frac{0{\cdot}33 \times 10{,}000 \times 300}{3{\cdot}27 \times 40} = 7{,}645.$$

The angle of twist is at all times very small, seldom exceeding 1·3° in 120 inches of length at full power, so that the pair of discs of a torsion meter are necessarily of large diameter to give accurate and fine readings, especially as in many ships it is possible to use only a very short length of shaft. When possible, however, it is certainly desirable to cover as long a portion of a tunnel shaft as possible.

The shafting of a ship, however, has another load on it besides that of transmission of the power generated by the engine, for the thrust block is usually close to the engine room, and consequently the whole of it from the screw to the block has to resist the thrust. This may amount to as much as to equal 20 per cent. of the torque force, but its influence on the twist is really very slight, probably only about 1·5 per cent. at most; generally it is so small as to be negligible, as stated by Prof. Hopkinson, from a considerable experience gained in testing his torsion meter. Each shaft should be and is tested by levers and weights to ascertain its actual resistance to torque, and to definitely determine the real and exact amount necessary to twist the shaft through a definite angle. From the observations so obtained, it is easy to make a diagram, to which reference may be made on trial when the torsion meter is showing the value of θ to obtain the corresponding S.H.P. per revolution, so that by merely multiplying by the number of revolutions per minute the full S.H.P. is determined.

It is, however, not sufficient for a marine engineer to know only the power developed by the engine or turbine; he must be acquainted equally well with the power taken and delivered at every critical point of the ship and machinery, so that he may make up a balance sheet which shall show on the one side the maximum gross power the engine has given out, and on the other side the disposal of the same, so that nothing is missing and unaccounted for. In fact, the economies of engineering are of the very highest importance, and cannot be too carefully studied by all concerned from the designer to the engineer in charge. Each and every part of the machinery of a ship must be in such a state as to be working at the highest rate of efficiency, and how to ascertain that rate exactly must be the constant care of those in charge of it and responsible for its good and economic working.

Thrust Horse-Power is an expression that will be more often used in the near future than in the past, inasmuch as means may be soon provided whereby the actual thrust on a line of screw shafting will be as easily shown as the pressure in the condenser now is. Inventors have for some time turned their attention to devise some simple and easy method of doing this, and one of them, Mr. Heck, fully disclosed to the members of the Institution of Naval Architects (*Transactions*, 1909) two or three such methods that are within the bounds of practical politics, while not quite satisfying engineers that they are the ones to be finally adopted. They are all ingenious and capable of giving results fairly free from inaccuracy. So far, these and some others are based on the principle of showing, by means of a hydraulic ram or its equivalent, the pressure per square inch in the chamber, or the total load on the equivalent ram, and how to make allowance for the friction of packings, etc. A more recent idea is to make the flange couplings hollow and elastic, fit them together, water-tight, and fill the space between them with water whose pressure is indicated by a gauge in the usual way. The thrust block might also be used to give its own indications of thrust (as also suggested by Mr. Heck), if it were mounted on roller bearings or suspended on a stirrup and its thrust taken by a pair of hydraulic rams with chambers connected, and the water in them acted on a gauge and spring or loaded resistance of some kind, as the hydraulic brakes on a gun carriage.

The Gross Power of a Steam Engine is, of course, that generated in the cylinders by the steam pressure on the pistons acting through the space traversed by them. This is known to the marine engineer as the **Indicated Horse-Power.**

Shaft Horse-Power is that transmitted through the tunnel shafting from the engine to the propeller, and is, therefore, the net product of the engine or the gross power less that absorbed in moving the engine and its appurtenances, called the **Friction Horse-Power.**

The mechanical efficiency of the engine is, therefore, S.H.P. ÷ I.H.P.

Brake Horse-Power is also that transmitted through the shaft to a resistance capable of absorbing it just as the propeller does that from the marine engine. In this case the brake not only takes the power, but indicates exactly the torque or twisting moment from which the B.H.P. may be calculated. Brake horse-power should and does coincide very closely with S.H.P.

The Thrust Horse-Power is that exerted by the screw in pushing the ship forward, and is measured by multiplying the actual thrust in pounds by the number of feet moved through by the ship in a minute and dividing by 33,000.

If S is the speed in knots per hour, and T the thrust in pounds,

$$\text{Thrust horse-power} = \frac{T \times S \times 6{,}080}{33{,}000 \times 60} = T \times S \div 326.$$

This measures the capacity of the screw as a propeller, consequently

The efficiency of the screw = T.H.P. ÷ S.H.P.

Indicated Thrust is an expression introduced by Dr. William Froude as a measure of the thrust of a propeller.

$$\text{Indicated thrust} = \frac{\text{I.H.P.} \times 33{,}000}{\text{pitch of screw} \times \text{revolutions}}.$$

Tow Rope Horse-Power is that necessary to propel the ship at the required speed if freed from what are called the augmented resistances, the chief of which is due to the action of the screw itself on the stern of the ship to retard her motion. If T_R is the tension on an imaginary tow rope from the ship to the towing agency, and S the speed of the ship in knots per hour, then

$$\text{Tow rope horse-power} = \frac{T_R \times S}{33{,}000} \times \frac{6{,}080}{60} = \frac{T_R \times S \times 101{\cdot}3}{33{,}000}.$$

This is the net-horse-power of propulsion; hence—

$$\text{Propulsive efficiency} = \frac{T_R \times S}{326} \div \text{I.H.P.}$$

Net Horse-Power can be calculated, as already shown, by estimating the skin resistance and the residuary resistance, adding them together, multiplying the sum by the feet passed through in a minute, and dividing the product by 33,000 lbs.

If Sr be the skin resistance and Rs be the residuary, both in pounds, and S the speed of the ship in knots per hour, then

$$\text{Net H.P.} = \frac{(Sr + Rs) \times 101{\cdot}3}{33{,}000}, \quad \text{or} \quad \frac{Sr + Rs}{326}.$$

Piston speed and revolutions enter largely into the calculation of horse-power, and, therefore, important factors at every stage of a design. The modern engineer does not permit himself to be hedged in and bound by the arbitrary rules of former generations, nor indeed would they have been to the extent they did had they had the benefits of the knowledge, experience, and materials now enjoyed. *Festina lente* was the policy of every progressive as of every prudent engineèr in early days, and may with advantage form a portion of that of modern ones, who now benefit by the failures as well as the successes of the pioneers of the profession. The original users of the screw propeller cannot be accused of fearing high revolution of that instrument, and had those who followed them so soon attempted to make engines which could revolve at the speeds of these screws when directly coupled to them, such wretchedly low efficiencies as exhibited in the machinery and propeller of the "Greyhound" under Dr. William Froude's analysis would not have been possible.

To-day both piston speeds and revolutions of engine are much higher than prevailed with the old compound engine. The better distribution of torque with triples and quadruples, and the balancing of the inertia forces have enabled this to be done successfully, so that to compete with the turbine still higher revolutions may be attempted in the future with engines specially designed for it.

Experience has shown that heavy marine pistons may run safely at a mean velocity of 900 feet per minute, and, in some instances, the pistons of some large vertical engines in first-class cruisers have reached a velocity of even 1,000 feet; while higher speeds still have been attained in torpedo-boat destroyers, whose pistons move, when run at express speed, at a velocity over 1,200 feet per minute. Although there is no difficulty in causing a piston to move at even higher speeds than these, it is doubtful if there would be any advantage in doing so, besides which the risk of causing serious damage to the cylinders, and precipitating a break-down without any warning, is very great. There is no doubt that a well-fitted piston, moving in a smooth and true cylinder at a speed of 1,000 feet per minute, will work well so long as the rubbing surfaces receive some lubrication from the moisture of the steam or the oil injected, and there is not the slightest fear of danger under these circumstances; but if, with a little priming, scum is carried into the cylinders and causes abrasion of the rubbing surfaces, an immense amount of mischief may be done in a few seconds. Moreover, when the cylinders wear a little out of shape from one cause or other, so that the packing-rings will have lateral motion, the danger increases with the velocity of the piston. In the Navy engines supplied with steam from water-tube boilers have no internal *oil* lubrication beyond what passes in on the rod surfaces; so that in destroyers, whose pistons are moving at a speed of 20 feet per second, there is only the moisture from the steam to lubricate them.

Although the revolutions of a screw engine may be, within certain limits, as few or as many as the designer chooses, experience or prejudice has fixed very closely in practice the limits beyond which it is not considered expedient to go. In the days of the geared engine, the screw revolved three or four times to one of the engine, and no objection was raised to the small screw and the high number of revolutions; later such a thing was deemed very objectionable—on the ground of excessive speed of piston and excessive friction in journals. The slow-moving engine was quoted as a proof of the economy of slow piston speed and small friction without being a real foundation for the argument.

The fine lines of the older steamships admitted of the small screw, which was the accompaniment of the engine, by necessity geared. Bluff ships, as now built for mercantile purposes, require a much larger screw for the same power of engine and dimensions of hull than formerly obtained; and it is not to the *slowness* of the pistons that they owe their economy, but rather to the *small number* of strokes per minute made by them in turning the large screw.

An engine requires a certain power to be expended in moving it through *one revolution* to overcome internal resistances; if the number of revolutions is 80 per minute, this power will be double that at 40, and, roughly, will vary directly with the revolutions. But the resistance of the propeller, caused by friction of the water on the surface of the blades, will increase roughly as the square of the revolutions, so that the *power* expended to overcome this resistance at 80 revolutions is eight times that required at 40 revolutions. If now the screw can be so altered with respect to pitch that, at 40 revolutions, the same speed of ship is obtained as at 80 revolutions, the indicated horse-power will be found to be considerably less; and although the *coal consumed per I.H.P.* will not be less, and may possibly be more than before, the consumption per day will be considerably less. Now, although this economy is co-existent with decreased piston speed, it is not due to it.

The object of a high rate of piston velocity is to decrease the piston area and that generally for the sake of reducing the size of the engine. But an increased velocity may be obtained either by increasing the stroke of piston, or by increasing the number of revolutions; if the former method is adopted there will be no decrease in the size of engine; but, on the contrary, an increase in space occupied and in the weight. If a high piston speed is obtained by a high number of revolutions, a smaller cylinder will suffice for a certain indicated horse-power than if the same piston speed were obtained by length of stroke alone. In other words, engines which are required to develop a certain power in a minute will vary in size of cylinder inversely as the number of revolutions per minute, all other things remaining constant; and if the cylinders are of the same diameter, the stroke will vary inversely as the number of revolutions.

The piston speed of many engines is governed entirely by circumstances beyond the immediate control or will of the designer. An example of this is the case of the paddle-wheel engine with vertical oscillating cylinders. If the position of the shaft is determined by the structural arrangements of the hull, as is often the case, then the diameter of the wheel is fixed, and the speed of ship fixes the number of revolutions to be made by the wheel; the length of stroke of piston is limited by the distance from the centre of the shaft to the floors or keelson of the ship. Further, if the engineer is free to decide the position of the shaft, any attempt to increase the piston speed by placing the shafting high is frustrated by the fact that, the higher the shaft the larger will be the wheel, and consequently the fewer the revolutions. If the engine is inclined, then the designer may fix the diameter of the wheel to suit the revolutions which he deems most advisable, or he may fix the position of shaft to suit the ship's structure, and still be free to choose the stroke of piston.

Again, the horizontal engine had to be designed so as to accommodate itself to the space allotted to it in the ship, which means that only a limited length of stroke was permissible. The revolutions, however, in this case could be varied considerably; but there is, after all, a limit to the number; beyond this limit any increase will result in very little gain in speed, and a very certain loss of efficiency. If the screw is of comparatively small diameter, owing to the shallow draught of the ship, a higher number of revolutions than usual is absolutely necessary to project a sufficient mass of water back to propel the ship forward with the necessary velocity; and it is the medium number, or that number at which the engine can be run without loss of efficiency so as to obtain the maximum speed of ship that is so difficult to decide, and which can only be determined with any degree of certainty by experiment.

One great feature which places the vertical engine so much above all the other forms of screw engine, as an economic and good working machine, is its superior length of stroke. Power for power, the vertical engine always has exceeded the horizontal in this respect; and although in the practice of the past there was no very great difference in this matter between the two types, the tendency is now to make the stroke as long as is possible or convenient in the engines of all ships.

The advantages of the long stroke are due to the corresponding decrease in piston area. Two engines of the same power, and working at the same number of revolutions, must have the same *volume* of cylinder; or, to speak more correctly, the pistons must sweep out the same volume if their efficiency is the same. The crank-shafts will be of the same diameter, and the crank-pins, also, practically of the same dimensions. Now the one with the long stroke will have smaller pistons than the other, consequently the total pressure on the pistons will be smaller—and, in fact, is inversely proportional to the stroke; consequently, the pressure on the guides, crank-pins, and journals will vary in the same way, and the friction on them correspond also. The lateral pressure of the piston packing rings will vary with the diameter, so that any reduction in diameter will produce a corresponding reduction in the friction.

But, perhaps, so far as economy in working is concerned, there is no more important consideration than the reduction in clearance space effected in the low-pressure cylinder by the reduction in piston area. The steam ports will be nearly the same in section, whether the engine be long or short stroke; but the space between the piston and cylinder-ends is very considerably reduced, and will vary inversely as the length of stroke, because the axial distance of piston from the cylinder-ends is constant.

The Rate of Revolution of Marine Engines* at full speed varies roughly inversely as the square root of the nominal horse-power, which for this purpose may be taken as N.H.P. = D × S ÷ K.

D is the diameter of the low-pressure cylinder (or the equivalent, if there are two) in inches.

S is the stroke of piston also in inches.

K is a coefficient of 15 for two-stage compound, 12·6 for triple-compound, and 10·5 for quadruple-compound engines. Then—

Rate of revolution per minute . . . $= Q \div \sqrt{\text{N.H.P.}}$
For the ordinary cargo boat, Q . . $= 1{,}200$.
For express steamships, Q . . . $= 1{,}800$.
For naval and very fast express ships, Q . $= 2{,}250$.

Example (*a*).—The proper rate of revolution for an express steamship having cylinders 30 inches, 45 inches, and 70 inches diameter × 42 inches stroke.

Here $$\text{N.H.P.} = 70 \times 42 \div 12{\cdot}6, \text{ or } 233.$$
$$\text{Revolutions per minute} = 1{,}800 \div \sqrt{233}, \text{ or } 118.$$

Example (*b*).—Rate of revolution for a warship having engines 33 inches, 52 inches, 64 inches, and 64 inches diameter × 48 inches stroke.

Here $$D = \sqrt{2 \times 64^2}, \text{ or } 90{\cdot}5.$$
$$\text{N.H.P.} = 90{\cdot}5 \times 48 \div 12{\cdot}6, \text{ or } 345.$$
$$\text{Revolutions per minute} = 2{,}250 \div \sqrt{345}, \text{ or } 121.$$

Revolutions.—Although there is a very considerable range for choice of number of revolutions of the engines of most merchant steamers, there are certain well-defined limits beyond which very few practical engineers go.

Very few screw engines are now worked below 75 revolutions per minute when in good condition; and it is at this speed that most of the engines of the large mail steamers are kept running on the voyage so long as the weather permits. The engines of warships, for two very good reasons, work at much higher speeds. Their machinery must be light, and go into a small space, so that it is necessary to make an engine of certain dimensions to suit these

* The N.E. Coast Inst. E. and S. recommend the following as the rule for rate of revolution of cargo-ship engines *when on voyage*:—

$$N = 32\,(S + 4) \div S \text{ or } \frac{128}{S} + 32.$$

conditions, and cause it to develop the requisite horse-power by running at a higher number of revolutions. The speed of a warship is much higher in proportion to its size than is that of the merchant ship, while the draught of water is no more, and often less. For these reasons the screw of the warship is small for the power to be developed, so that even if large engines were admissible to drive the screw, they would be of small advantage, as they would have to move at a high rate. It will be seen, then, that small fast-running engines are a necessity, and especially is this so with modern warships, whether armoured or unarmoured. The latter must be as fine as possible, and every ton of weight saved to obtain the very high speeds which their service demands; the former demands every sacrifice to save weight in machinery, for the sake of adding it to the armour and armament.

Since a warship has so seldom to steam at full speed, and when she does, it is only for a short period, the short-stroke fast-running engine is not so very objectionable, and rigid economy is quite a secondary consideration in war questions.

The following table gives the number of revolutions at which naval engines are run on their trial trips:—

TABLE XXIXa.—Rates of Revolutions of Screw Propellers.

Type of Engine.	Description of Ship.	Power of each Engine to a Screw.	Revs. per Minute.
Reciprocators,	Battleships and 1st-class cruisers, . . .	19,500 to 15,000 I.H.P.	130 to 140
,,	2nd-class cruisers, .	4,500 to 6,500 ,,	140 to 146
,,	3rd-class cruisers, . .	3,500 to 5,000 ,,	225 to 245
,,	Scouts, ...	7,000 to 8,500 ,,	210 to 202
,,	Destroyers, ...	2,000 to 3,500 ,,	360 to 400
,,	Torpedo boats.	..	..
,,	Large Atlantic expresses,	15,000 to 20,000 ,,	76 to 80
,,	Large mail steamers, .	5,000 to 15,000 ,,	90 to 80
,,	Large cross-channel steamers, ...	3,000 to 4,000 ,,	175 to 150
,,	Small cross-channel steamers, ...	2,000 to 3,000 ,,	200 to 175
,,	Large twin-screw cargo steamers, . .	2,200 to 4,500 ,,	80 to 90
,,	Large single-screw cargo steamers, . . .	2,500 to 4,500 ,,	80 to 70
,,	Medium single-screw cargo steamers, . . .	1,000 to 2,500 ,,	90 to 80
,,	Yachts and small craft, .	250 to 500 ,,	150 to 250
Turbines,	Battleships, . . .	5,500 to 7,500 S.H.P.	320 to 300
,,	Large cruisers, . .	10,000 to 20,000 ,,	200 to 175
,,	Second-class cruisers, .	6,000 to 12,000 ,,	500
,,	Scouts, ...	4,500 to 7,500 ,,	750 to 600
,,	Destroyers, . . .	2,500 to 5,000 ,,	940 to 750
,,	Atlantic expresses, largest,	15,000 to 17,500 ,,	190 to 165
,,	Express steamers, large,	4,000 to 6,000 ,,	300 to 190
,,	Cross-channel, large, .	3,000 to 5,000 ,,	600 to 450
,,	Yachts and small craft, .	1,000 to 2,000 ,,	1,000 to 750
,, geared,	Expresses, . . .	2,500 to 3,000 ,,	300

The stroke of horizontal engines varied from 18 inches of the gunboat to 54 inches of the large armour-clad, and the vertical engines of the Navy vary from 18 inches in the "destroyers" to 51 inches in the first-class cruisers and battleships. Latterly, with the four-cylinder engines, the stroke has been 42 to 48 inches in the large ships.

Length of Stroke.—For very many years there existed a standard scale for the stroke of the vertical engine of the mercantile marine, and although there was no written law which guided engineers in the choice of this important dimension, it was so well known that only *the diameter* of the cylinders was mentioned in speaking of the size of engine, and in most of the rules for nominal horse-power used by manufacturing engineers in their dealings with shipowners, no direct allowance was made for length of stroke. With the walls of the cylinders only jacketed the best relation of diameter to stroke was as 1 to 1·5. So the ratio of stroke to diameter of H.P. cylinder is usually 1·7 and the ratio of stroke to diameter of L.P. cylinder 0·9 to 0·6.

The following Table gives the stroke corresponding to the different powers; the one column giving the standard, and the other the stroke as existing in ordinary every-day practice in the mercantile marine :—

TABLE XXIXb.

N.H.P.	Standard Stroke.	Stroke, as in common practice.	N.H.P.	Standard Stroke.	Stroke, as in common practice.
20	12 ins.	15 ins. to 18 ins.	140	33 ins.	33 ins. to 42 ins.
30	15 ins.	18 ins. to 21 ins.	160	33 ins.	33 ins. to 42 ins.
40	18 ins.	20 ins. to 24 ins.	180	36 ins.	36 ins. to 45 ins.
50	21 ins.	24 ins. to 30 ins.	200	36 ins.	36 ins. to 48 ins.
60	24 ins.	24 ins. to 27 ins.	250	39 ins.	39 ins. to 54 ins.
80	27 ins.	27 ins. to 30 ins.	300	42 ins.	42 ins. to 54 ins.
100	30 ins.	30 ins. to 36 ins.	400	45 ins.	45 ins. to 60 ins.
120	30 ins.	30 ins. to 39 ins.	500	48 ins.	48 ins. to 66 ins.

TABLE XXIXc.—Revolutions and Piston Speeds according to Rules recommended by N.E. Coast Inst. E. and S. for Cargo Steamers on Service.

Stroke	Revolutions.	Piston Speed.	Stroke.	Revolutions.	Piston Speed
Feet.	Per Minute.	Feet	Feet.	Per Minute	Feet.
1·50	117	352	3·25	72	464
1·75	105	368	3·50	69	480
2·00	96	384	3·75	66	496
2·25	89	400	4·00	64	512
2·50	83	416	4·25	62	528
2·75	78	432	4·50	60	544
3·00	75	448	4·75	59	560

CHAPTER IX.

GENERAL DESIGN AND THE INFLUENCES WHICH AFFECT IT.

The General Design and Arrangements of Marine Engines are to-day characterised by simplicity perhaps more than anything else, notwithstanding that multiplicity of parts and connections gives the modern engine-room an air of complexity that was wanting in the older ships. Forty years ago there were, as a rule, only two cylinders to the main engines: both air and circulating pumps, as well as the feed and bilge pumps, were worked from the pistons, direct in the Navy, and in that way or by means of levers interposed in the mercantile marine. There was, as a rule, only one auxiliary feed pump, and in large ships one fire engine in addition, which did duty for a bilge pump as well as for deck service when required; merchant cargo ships with ballast tanks had also the second auxiliary pump for emptying them, and arranged to act for general purposes. Steam starting gears were being used generally with very large engines, but never with small ones. Steam steering gears were equally rare, and there was, of course, no electric light engine, nor any refrigerating plant. Distillers had been introduced for giving fresh water as auxiliary feed to the boilers by Hall, of condenser fame, thirty years before this period, but the idea had not taken on. There were, however, in general use the well-known Normandy distillers for producing drinking water; every Naval ship had a set, and they were to be found in the better-class passenger steamer. The mercantile engineer had, for the exercise of his talents and the occupation of spare time, the care of the steam winches, just as the naval engineer had in some ships the charge of the turret and turntable engines and ammunition hoists, etc., outside his own domain. Very few ships had more than one main engine or a steam launch or pinnace, and no ship had air compressors. All these things have been added one after another, until the care and anxiety of the chief engineer is no longer centred and concentrated on the main engines, but now it is rather the very numerous parasites of them and of the crowd of outside machinery—machinery so necessary to the well-being of the modern warship and express steamer—that gives him the most concern. But, in spite of all this, there is about the main engines, especially if they be turbines, a simplicity that is commendable under such circumstances. The day for fancy design and odd arrangements of machinery has gone; in their place there is now only a stereotyped system that marks the final stage of natural selection and the survival of the fittest. The engine of one modern engine builders differs very little in essence from that of another; the turbine, it is true, has its variation in principle as well as in arrangement, but it, too, is gravitating to the stereotyped design, although makers may still each have his own, differing in form of details and fittings.

The accepted and approved type of reciprocating engine is, of course, the vertical inverted direct-acting one, whether it be worked by steam or by the internal combustion of oil or gas. The cylinders are placed in line each over its own crank, and, as a rule, they are separated one from another; each has its columns braced and secured so as not to be dependent on its neighbour for stability or steadiness. The connections for steam transmission are by pipes so fitted as to expand and contract freely under thermal or pressure forces without acting or reacting on the cylinders and so cause them bad alignment. At the same time the *general structure* of columns is braced together, so as to give it, as a whole, additional stability against external disturbance, as from the shock or strain of rolling, pitching, and collisions. Small mercantile engines, however, are still made with the cylinders cast or bolted together in pairs, but the tendency is in the direction of separating all of them, especially when the stroke of piston and consequently their length is greater than that usual with the high-speed land engines of equal power. Each cylinder now operates on a single complete piece of crank shaft running in two bearings coupled to the next cranks; this is so in all but very small engines (under 100 N.H.P.) or in larger engines, where weight and space have to be cut down to the minimum, when the shaft is then in one piece. The valves are now seldom placed on the cylinder outer sides, but generally on the fore or aft side, so that the valve spindle is immediately over the shaft, and consequently capable of being driven by the double eccentrics and link motion direct without the interposition of weigh shafts and levers. Steam reversing gears are fitted to all but the very smallest engines, and are generally of the push-and-pull type introduced and still supplied by Brown Bros., of Edinburgh, for the mercantile marine, while the "all round" worm and wheel gear is most frequently used in the Navy. The former is decidedly the handier, and can now be obtained at quite as low a cost as the all round gear, but the latter can be used to turn the main engines. Similar push-and-pull gears, of various designs, are employed to operate the very large stop valves of the reciprocating and turbine engines, as they do also the heavy change valves of the latter, whereby the turbine is reversed by changing the incoming steam from the head-going to the stern-going part of it, or when the exhaust steam from the L.P. cylinder is diverted to a L.P. turbine. The circulating pump is now, in all but the small mercantile engines, of the centrifugal type driven by an independent engine; in all sorts of fast-running engines the air pump is also, as a rule, absent, and it, too, has its separate engine. With turbines where high vacuum is a necessity for high economy of steam, the double-stage air pump of Mr. Weir is often employed; there is still a prospect of the centrifugal pump being employed in series on this service, as it has been used for boiler feeding and delivering water generally against a high "head" on land.

The Admiralty long ago initiated, and the mercantile marine followed in express steamers, the practice of removing the whole of the feed and bilge pumps from connection with the main engines. The merchant ship, however, was really first in the employment of the independent feed pump with automatic regulating gear, whereby its speed was only such as to give just the necessary supply of feed demanded by the stokehold watch-keeper. Now, in all important steamers, and certainly in all turbine-driven ships, the feed bilge fire and general service pumps are absolutely separate and independent of one another; moreover, they are generally in duplicate to avoid the risk of stoppage of the main engines in case of the failure of an auxiliary.

Formerly the Condenser was a huge cast-iron vessel, forming an integral and important part of the engine structure or framing, to which it generally gave a massive foundation for the guide columns, and contributed to the stability of the framing (*v.* fig. 71), as well as a firm support for the bearings of the pump beam's weigh shaft. With the advent of the three-crank engine came the tendency to separate the condenser from the engine frames, and as this kind of engine became lengthened out, and; furthermore, when there came a four-crank engine to stay, it was no longer expedient to have it with such long tubes as would be entailed with a condenser body the whole length of the engine. Moreover, the addition of three and four sets of columns or column feet on it made it a costly and somewhat risky casting, as well as a heavy portion of the weight of the machinery. To-day the condenser is generally a distinct and separate vessel placed close to the L.P. cylinders, and of a form suitable to its own particular service, and in no way subservient to any other such service as was formerly exacted from it when it formed part of the engine superstructure (*v.* fig. 68). The cylindrical form was the common one, on account of its natural strength and cheapness, but now the demand of condenser experts have caused the cylindrical to be rejected for one with the heart shape cross-section, notwithstanding that the general requirements of such experts can be carried out sufficiently well in the cylindrical shell by fitting it with the form of special baffle directors (*v.* fig. 116), or in the rectangular contraflo design of Mr. Morison (*v.* fig. 117).

The framework of the engine is now much as was usual formerly in its general design, but greater care is exercised to give both general and local stiffness to every part, which, no doubt, is an improvement which tends to the better working of all engines in every way.

The Lubricating Arrangements are much more extensive and complete nowadays than formerly obtained, and the necessity for a regular steady and positive supply of oil to every bearing guide and pin is now recognised and provided by means of small pumps worked by the engines, or from overhead tanks, which deliver a steady stream at a pressure high enough to force the lubricant into every part requiring it.

The Design of an Engine is influenced by External Causes, which are not always seen or even acknowledged, but nevertheless are often in active operation all the time. There is even in the engineering and shipowning world also that which is generally elsewhere called fashion, exercising its powers, and imposing on the designer conditions with which it is often futile to struggle. Even the better general and technical knowledge possessed by experts to-day does not always preserve them from following a lead which is more or less of a blind nature, and from starting out in directions quite contrary to the convictions based on their experience, and for no other reason apparently than because someone else has done so with all the appearance of success. The trial and error system still prevails largely, in spite of the technical and scientific training now so freely obtainable everywhere at quite low cost. Model experiments are always interesting, often instructive, and, when properly understood, may be good and safe guides for directing the proceedings of the work-day engineer, but such models, even when telling truth, do not always tell *the whole* truth; what they suppress often comes out as a ghastly truth when the full-sized working engine is produced; what was a trifling matter and hardly tangible in the miniature becomes in the great a

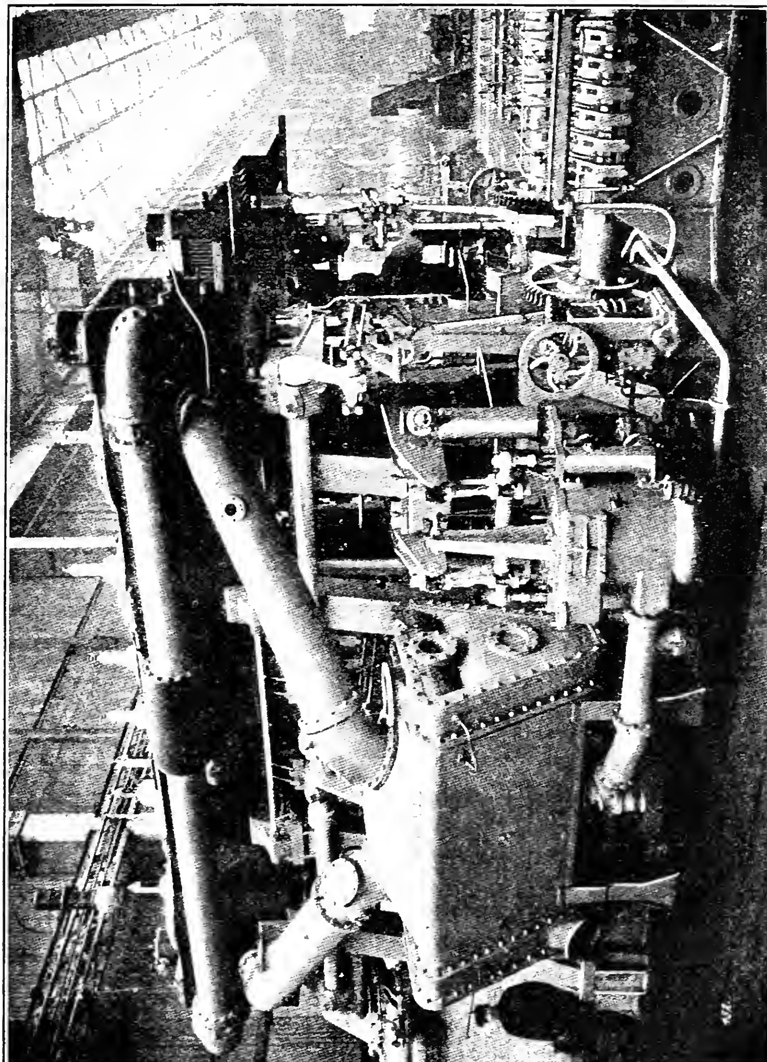


Fig. 68.—"Weir-Uniflux" Condenser, as fitted to a Four-cylinder Triple-expansion Engine.

terrible incubus, producing the most damning consequences; what was a pigmy in the small engine and quite easy of control becomes a very Frankenstein in the large one, irreducible, and carrying all before it, and spoiling everything. It behoves everyone, therefore, to use models and model experiments with great discretion.

The Supply of Materials exercises a powerful influence on the designer of engines generally, but especially on him who has to cut down to a minimum the weight of and space occupied by machinery in the way the marine engine builder is compelled to do. When Brunel designed the "Great Britain" there was no steam hammer in existence and no forge capable of making so large a shaft as that required for the ship; consequently he had to be content with one of cast iron when at that stage; fortunately for him, Nasmyth invented just then the steam hammer, and made one in time to produce a wrought-iron shaft, with which the first voyage of that ship was made. Cast iron was largely used for all parts of marine engines, and designs were made accordingly. When copper was boomed up to £80 per ton* and even higher by an enterprising but somewhat short-sighted syndicate, the equally enterprising and versatile marine engineer adopted steel and iron for piping, and had other things made of cast steel which formerly had been exclusively of bronze. Steel castings, doubtful and often unsatisfactory as they may be, have served their turn in changing the design of many parts of an engine, but what has effected the greatest departure from old practices to that which prevails to-day is the cheap and unlimited supply of excellent mild wrought steel. By its means the pressure of steam possible in cylindrical or tank boilers has been raised from 100 to 240 lbs. per square inch, and their possible diameter increased from about 14 feet to 18, while their cost has been reduced very considerably. The internal plates of such boilers used formerly to be made of "Lowmoor quality," at a cost of £27 per ton and upwards; similar plates in the superior metal (mild steel) and of very large sizes can be purchased now for a third of that sum; shell plates of iron, $1\frac{1}{4}$ inches thick, could be bought, but they were narrow and not very long; moreover, if of the full area the mill could turn out, they were very costly. Now, plates up to 50 feet long and 12·5 feet wide can be rolled, up to a thickness of 1·8 inches; shell plates are often $1\frac{3}{4}$ inches thick and 30 feet long, so that one, or at most two, plates when single-ended are sufficient; circular-end plates are made up to 13 feet in diameter. Forgings of every description and size can be made of this material, and, if needed, steel of a higher tensile, say 40 tons ultimate, with quite a good amount of stretch, can be obtained also in large size and at moderate cost; for special purposes at costs by no means prohibitive vanadium, nickel, or other high-class steels are made, and supplied of a quality and fitness beyond reproach.

Rolled bars of excellent steel can be obtained of any diameter up to 15 inches, and square bars to 6 inches; rectangular section bars can also be had in various sizes up to 15 inches by 2 inches, and 11 inches by $2\frac{1}{2}$ inches, so that steel caps for bearings, rod ends and other similar purposes can be made in wrought steel at a cost very little in excess of that of castings in iron. With the self-hardening tool steel and high-speed lathes and machine tools now in use, these rolled bars can be converted into bright ones at a trifling extra cost. The framework of an engine in wrought columns and tie bars is, therefore, now a much less expensive luxury than it was formerly, even for large engines. Then, too, the other sectional steel now obtainable in such variety has permitted designers to fashion engine beds and their framing in ways not possible without it, whereby great saving in weight is effected, and inasmuch as little

* Now £150, due to war conditions.

or no pattern-making is required for such designs, it is an economic method when the engine required is a special one with few or no repetitions of it expected.

The method of obtaining sectional material (introduced by Mr. Dick) by the extrusion of the zinc bronzes through dies has permitted of multiplying varieties without necessitating the expense involved in cutting rolls. This and the other ways in which these high-class tough, strong bronzes may be used have had no small influence also on the design of the smaller special engines.

Such things permit of refinements in design not possible in the days when engineers were limited to the choice of wrought iron of 20 tons ultimate tensile strength, or, by paying a high price, 25 tons at most. Crank and straight shafts of almost any size can be forged and machined at prices that were impossible for even small ones a few years ago. Even cast iron has been improved by selection and mixing, that without paying fancy prices for any of the good brands, the tensile and bending tests of castings are equal to the very best given by Fairbairn & Whitworth. Aluminium has not yet seriously influenced the marine engine designer; that it will in the near future is certain; its lightness alone will attract him, and probably alloys will be found which, while not adding seriously to their weight, will improve their strength and resistance to corrosion.

The Influence of Tonnage Laws on the hulls of ships is well known and obvious, but it is not limited to them. It pervades the whole of a ship more or less, and perhaps more so the machinery space than elsewhere. Latterly that influence has been more potent and insidious in its effect on marine engine design than believed to be possible years ago. Steamships have had from the earliest days of their construction some consideration given for the disadvantages under which they were worked compared with the sailing ship. The space occupied by the machinery has always been deducted from the gross tonnage, inasmuch as it could take no cargo; later on, in order to encourage shipowners in making the spaces, not only habitable, but healthful for those in charge of, and those labouring at the machinery, special allowances were made for the light and air spaces to engine and boiler rooms, as well as for those rooms themselves. If the actual total of the spaces which are allowed off the gross tonnage of a ship amount to 13 per cent. of it, the *actual deduction* permitted, so as to arrive at the net or register tonnage is no less than 32 per cent. In the days of low freights, costly fuel, etc., it is highly necessary to keep the register tonnage down, as on it the ship is taxed. Now, whereas in Naval ships the space allotted to machinery is always small for it, as indeed it was formerly only too often the case in the mercantile marine, until it was found that the enlarging of it enabled a considerable saving in working expenses to be effected at little cost and with inconsiderable drawbacks. Formerly with the two-cylinder engine often of quite small power, but requiring large supplies of fuel, it was quite impossible to arrange for a reduction of tonnage measurement so large as 13 per cent. without a much too serious sacrifice; nor, indeed, did then the exigencies of the times press for such drastic measures to obtain it, consequently it was only tug boats and express steamers having very large engines in proportion to their size that enjoyed these liberal concessions, as indeed they alone were intended so to do; consequently to the majority of steamers there was no gain by making the engine and boiler rooms larger than required by the bare necessities of the case, except that, until the amendment of the Merchant Shipping Acts, light and air spaces were not measured into gross tonnage, although they were included in the deductions from it; consequently every ton of space devoted to that purpose caused a virtual reduction of two tons in the register tonnage.

Since the space occupied by the machinery could not be laden with cargo, every foot of it was a foot less space in the cargo holds, it was made as small as possible, and the engineer designer had to exercise his wit to devise the most compact engine to occupy the least possible space. Hence the popularity with shipowners of the single-crank engine, either with one cylinder or two compound ones tandem; of the diagonal engine with a cylinder in each wing operating on a single crank (fig. 69), or with a third cylinder vertical, as in fig. 70. The compound engine with one cylinder vertical and the other horizontal, both operating on the same crank, found favour, as did also the engine with one pair above them and the other pair of cylinders behind operating on the same pair of cranks with levers, as revived by Mr. M'Alpine, latterly with the idea to provide a naturally balanced engine. Besides these typical instances, there were other ways in which the engine was treated by the ingenious, as, for example, in the way of special valve gears, etc., to reduce still more thereby the space occupied by it (fig. 71).

A further effect of the tonnage law, however, was more serious, inasmuch as it retarded the adoption of the better forms of triple- and quadruple-compound engines, and encouraged the manufacture of those that were worse; the fitting of valve gearing of almost fantastic design, which displayed an exaggerated inventiveness, while it gave endless trouble and anxiety to those in charge of it, was entirely due to the desire to save space and reduce the size of the engine-rooms. The early three-crank triple engines were often treated in this way; by shipowners they were objected to, although only too anxious to benefit by the three-stage system, on the ground that they required too much room; by the engine builder they were subject to such cutting and squeezing that the crank bearings and pins were reduced past the just minimum, and the valve gears were obstructions, and prevented the proper attention necessary to such working parts; moreover, they required such an expenditure of oil as to detract from the economy of fuel (*v.* fig. 71).

The two-crank triple, which in its way worked and did good service, was not so good an engine mechanically as the three, nor was the two-crank quadruple so good as the present four-crank one, although it was better in some ways than the two-crank triple. It was, however, due probably to the tonnage question that the quadruple engine came so soon as it did into the field of practical engineering, inasmuch as the claim that won it most attention and patronage at the first was the small space the two-crank engine occupied compared with that required for the three-crank compound. To-day all this is changed; every ship must have the 32 per cent. reduction; and, further, in some cases, since the law says that where the spaces which may be deducted amount to over 13 per cent. of the gross, the rate of reduction shall be one and three-quarter times the actual amount. In some very fully-powered ships the actual spaces may amount to even 54 per cent. of the gross; then the net or register tonnage is really only 5·5 per cent. of the gross. There are, however,

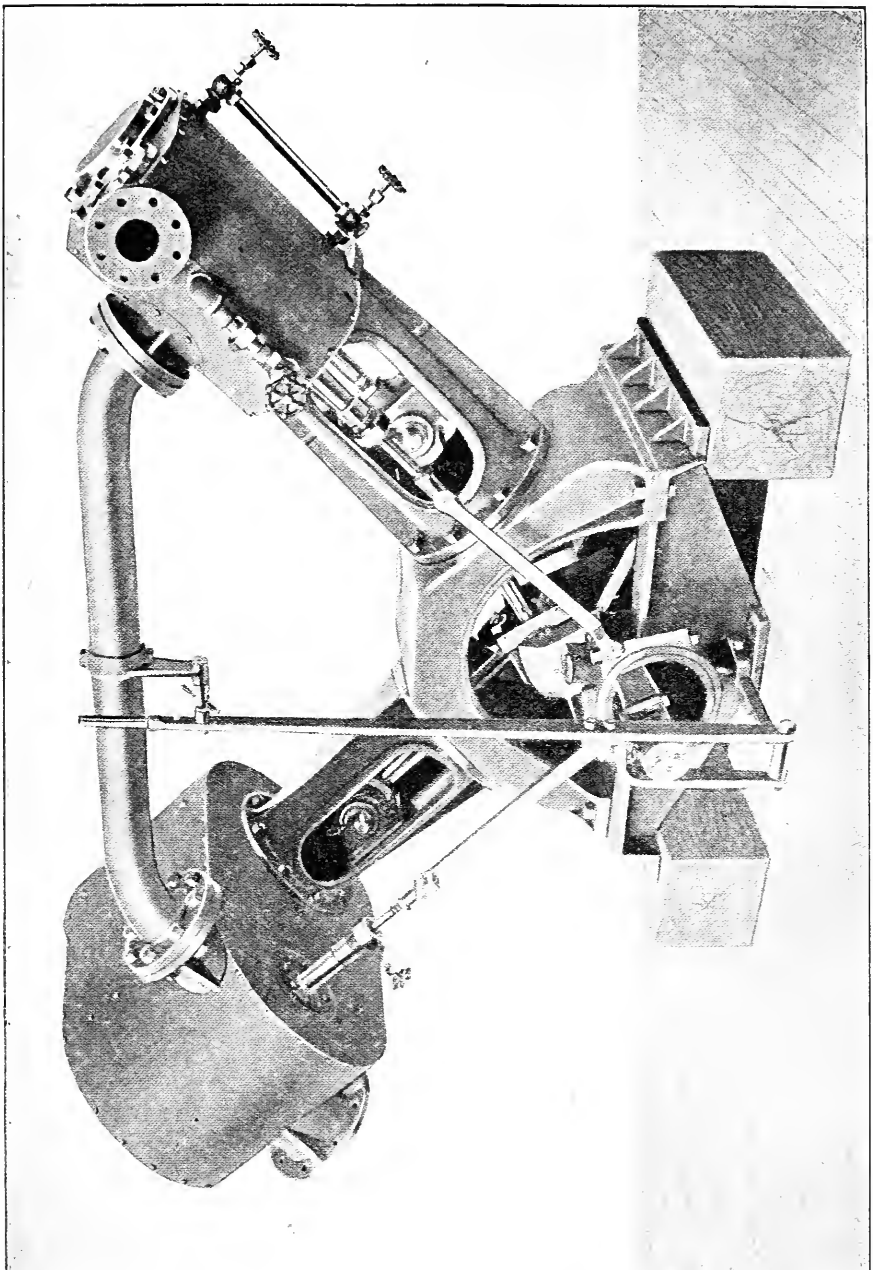


Fig. 69.—Single-crank Compound Engine. Two Cylinders, with one eccentric.
(Charles Ward, Charleston, U.S.A.)

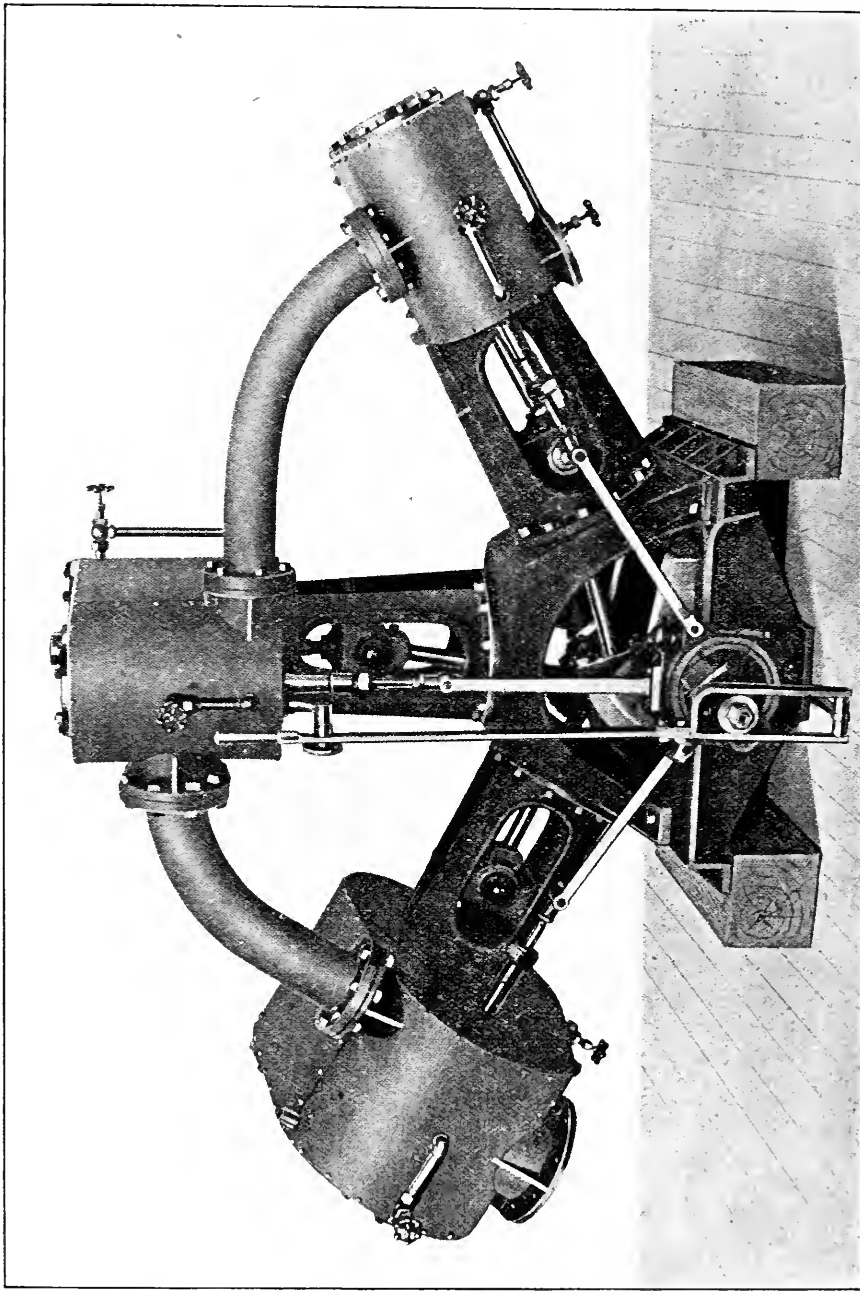


Fig. 70.—Single-crank Triple-expansion Engine.
(Charles Ward, Charleston, U.S.A.)

limitations set now so that the allowance may not amount to a public scandal, as it did quite recently to the Dock Companies, Harbour Boards, etc. Now it would seem as if an engine-room could not be too large, but, since the Board

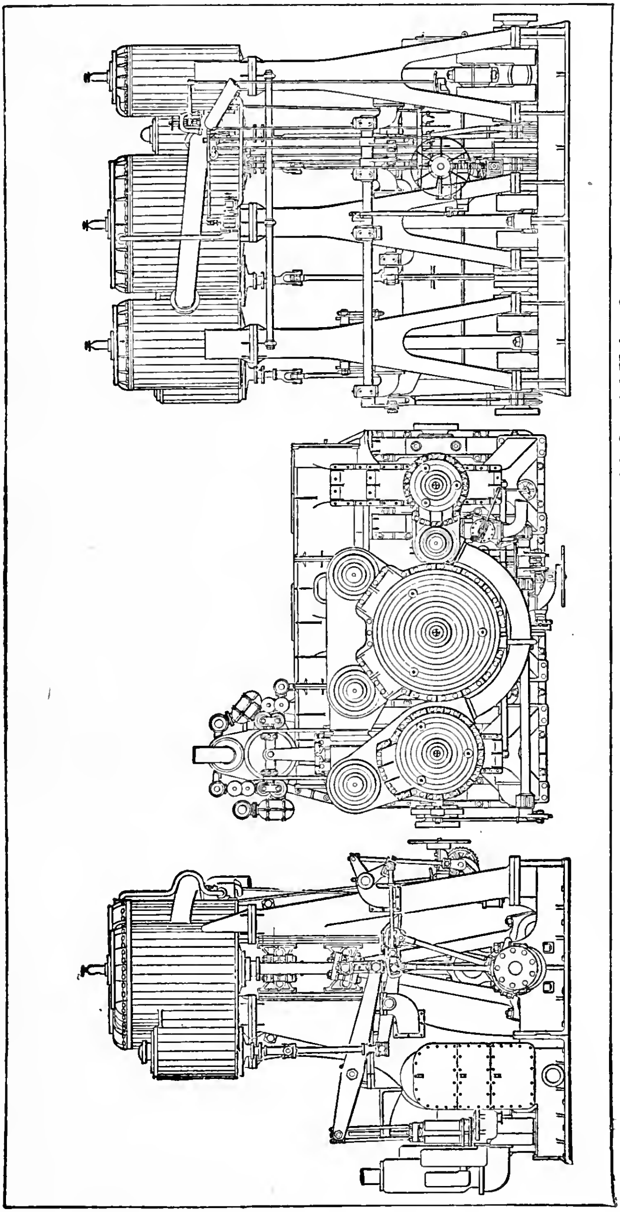
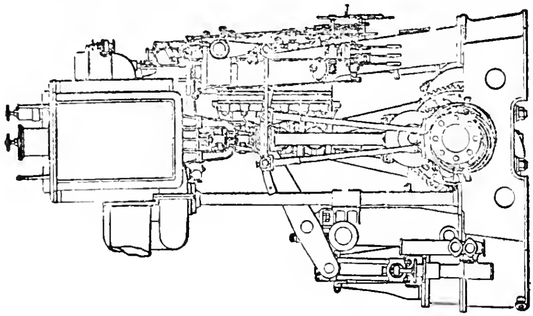


Fig. 71.—Three-crank Triple-expansion Engine, with Special Valve Gear.

of Trade will not permit it to be wholly measured off tonnage unless the size of the engines themselves will warrant the space allotted, there is a direct premium existing for making an engine as long as possible, and also to extend it athwartship liberally likewise. Consequently the four-crank engine (fig.



DIMENSIONS OF TRIPLE-EXPANSION FOUR-CRANK ENGINES

High-Pressure Cylinder	... ...	23½ In.
Intermediate	... ...	38 "
Two Low-Pressure Cylinders	...	43 "
Stroke	... ...	36 "

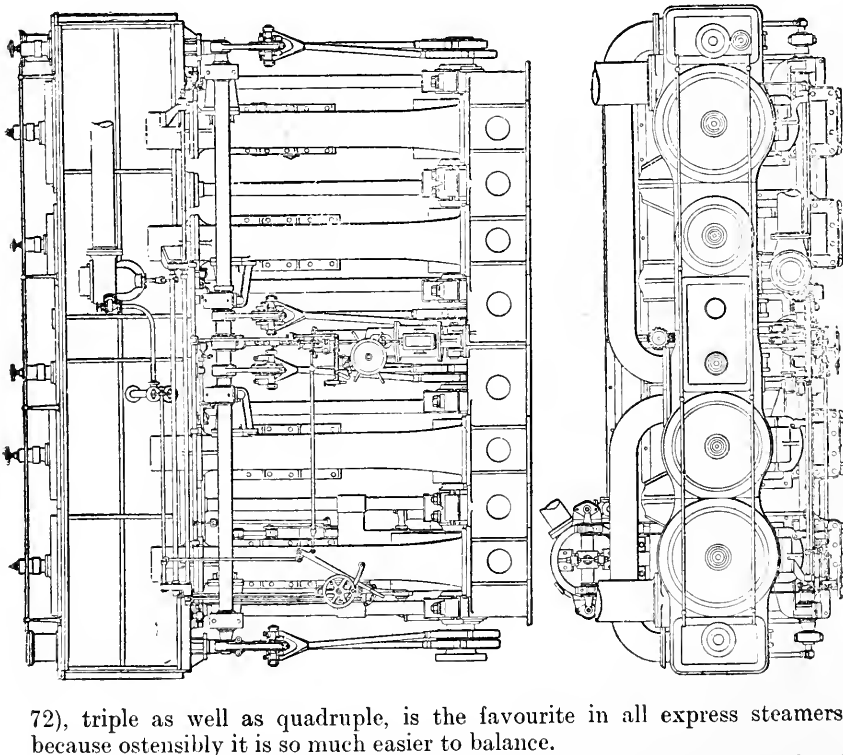


Fig. 72.—Four-crank Triple-expansion Engine Balanced on the Yarrow-Schlick-Tweedy System.

72), triple as well as quadruple, is the favourite in all express steamers, because ostensibly it is so much easier to balance.

The "Joys," "Marshall," and other special gear for driving valves placed in front or rear of the cylinders are consequently no longer necessary and so gone, and in their place the old and well-tried pair of eccentrics with their link motion is revived in their best forms, so that they are now with ample bearing surfaces, sufficient breadth of strap, etc.; they are, moreover, quite accessible. The condenser, for perhaps the same reason, sometimes is placed in the wings and the pumps of various kinds scattered about, each with its allotted and ample space. No longer are they and all other auxiliary engines and machines squeezed into odd corners and niches, which were narrow and inaccessible often, and always cramped and confined.

All this may not be altogether the result of the tonnage laws alone, but it is pretty certain that if there were not such good allowances off gross tonnage there would not be such liberal spaces, and if there were not roomy spaces, then the engines would have to be cut to fit them, such as they might be.

The Influence of the Board of Trade on marine engine design and practice in other directions has been, and is still, very great and, on the whole, beneficial. The better construction and design of the cylindrical boiler was largely due to the action taken by Mr. Thomas Traill and his staff officials of the Board about 1873 in formulating Rules and Regulations, which, while being somewhat arbitrary and often unnecessarily rigid in application, were, nevertheless, one of the chief means whereby the marine boiler has been so free from accident, slight and serious, for so long. The investigations carried out by the late Mr. Peter Sampson when assistant to Mr. Traill brought to light an enormous amount of useful information for the guidance of engineers generally, as well as for their own, in drawing up the Rules. That they then viewed all steel with a suspicion that did not seem warranted was the subject of much regret by those who felt sure that good steel could and would be produced at a cost which must, in course of time, drive wrought iron out of the market, if it was given the same freedom as accorded to wrought iron of every make and sort. Nevertheless, it must be admitted now that in thus keeping all steel under strict surveillance, much that was bad was prevented from coming into general use, and all makers of the material were thereby compelled to exercise the greater care in manufacture and treatment, which now permits of the greater freedom in its use. At the same time, there has been, and still is, though in a lesser degree, the regret that more encouragement is not given to those manufacturers who honestly strive to provide good and safe material for engineers' general use, having virtues superior to those of the common sort. In a general way, perhaps, low tensile steel is safer than the higher kinds, but it does not, therefore, follow that no high tensile steel can command the same confidence reposed in it that obtains with the "best mild steel." In this respect their progress has been slow in the mercantile marine. The high tensile steels are employed very sparingly compared with what might have been the case had the restrictions on their use been exercised in accordance with their real merits, instead of in compliance with the policy of the "Board." It is, of course, freely admitted that a Government Department has to be very discreet in its actions when dealing with one manufacturer and another, especially seeing that the desire is generally of all concerned for the practice by all officials to be uniform and absolutely impartial; that the treatment of all shall be on the same lines without differentiation, baseless or otherwise. In the use of an old and tried material like cast iron, however, there are not the same conflicting causes for singular treatment by the Board of Trade and its officials; yet, while at one time nearly the whole of the engine was made of cast iron, and some of the most important parts are still formed of it, where it could be squeezed out, it has been by the policy of both Admiralty and Board of Trade of late years to do so. Yet this material has some distinct virtues which render it not only a convenient but a safe one for the composition of certain parts. Under tensile stress it stretches more than steel does up to the elastic limit; to resist compression it has no equal, so that some structures which are liable when under load to be subject to a considerable amount of compression are really better made of it than of steel. Notwithstanding this, however, the tendency is to avoid its use, largely due to the policy and rules of these Government Departments on the ground of its comparative brittleness and liability to crack. The continued testing by expensive methods of the steel material, both in the cast, forged, and rolled state, as now used in engine construction, has tended to retard the use of it for a considerable period after steel makers had found the means of producing it cheaper than wrought iron. Since very little, if any, reduction in scantlings was claimed by engineers when substituting steel for wrought iron in many parts, it was of little or no consequence what its *ultimate* tensile strength was; it was, however, important, and is still necessary, to be assured that it is *tough and ductile.* This could be quite certainly and satisfactorily ascertained by submitting samples to a simple cold-bending test, and further to punching and upsetting tests, such as applied to rivets. Had such simple and inexpensive means been adopted, this superior material might have taken the place of common wrought iron, and even of cast iron, in many places long ago.

The restrictions of Government Departments, as a rule, tend to stagnation in engineering practice, while fulfilling the function for which they exist to protect the public from the dangers that might arise if greater freedom were given to engine builders. But neither at the Admiralty nor at the Board of Trade is there now raised that dead wall to rational and desirable progress; indeed, the Admiralty during the past two decades have become almost too progressive for the economic working of those who serve them.

The Balancing of Engines and Avoidance of Vibration is a distinguishing feature in latter-day marine engineering practice, and the advent of the turbine on shipboard has rather accentuated the necessity for the exercise of the art of balancing than abrogated it; for, although that instrument itself is free from the inertia defects of the reciprocating engine, it requires nevertheless the most careful of balancing itself; but to compete with the turbine-driven steamer, that one having reciprocating engines must have them so that they run with an absence of vibration as far as is possible. Thanks to that veteran engineer and practical scientist, Dr. Otto Schlick, who gave us so freely the benefit of his years of patient labour and careful research, we understand now, as we did not before, the reason for and the causes of all vibration in a steamship, and, better still, he had devised means for correctly ascertaining them and curing their effects, for which we were more indebted to him. Formerly it was not uncommon to attribute all the vibration of a screw ship to the action of the propeller; that some of it might be due to the momentum of the moving parts of the engines was, of course, apparent to all engineers having even an elementary knowledge of dynamics; attempts of a feeble and tentative kind were made to check them by fitting balance weights to the shafts opposite the cranks of the horizontal engine, or by casting with the turning wheels balance weights of segmental form. No doubt the inertia effects of the pistons and rods were to some extent neutralised by these balance weights, but it would appear that what the makers of these engines really aimed at was merely to balance the *weight* of the crank-pin, arms, and connecting-rod ends rather than the horizontal forces due to the movement of the pistons, etc. Balance weights were very seldom fitted to vertical engines, notwithstanding that from these statical considerations they were really the more needed in them than in the horizontal.

Nowadays balance weights are freely used in all classes of engine and for all sizes, but the application of them is effected with more discretion and judgment than formerly prevailed, so that much better effects are obtained with considerably less material and cost. The four-crank engine, balanced on the system introduced by Dr. Schlick and perfected in this country by Messrs. Tweedy and Yarrow, is without added weights, and now in general use; an almost perfect balance with an absence of vibration is obtained by this system with the special arrangement of the angles of cranks to suit the momenta of the moving parts.

Whether the engine be a three-, four-, or five-crank one, it must be satisfactorily balanced in every warship and express passenger steamer, and even in the cargo steamers, which may and often do convey passengers, it is desirable to avoid unnecessary vibration.

That Vibration arises in part from the Action of the Screws is only too true now as it was formerly, for the same causes exist in full force to-day notwithstanding our better knowledge of the subject, thanks to Dr. Schlick, for, after the engines have been most carefully and perfectly balanced, there often is manifest a residual vibration or trembling, which, while it may be quite local and not general, is nevertheless disagreeable. Moreover, the same defects are observable in as pronounced a manner in turbine-driven steamers notwithstanding the uniformity of torque and absence of any unbalanced inertia forces.

There are various ways in which a screw may set up vibrations, all of which, even the smallest, is capable of creating the evil if its application is intermittent and regular and its periodicity coincides with that of the natural vibration of the ship as a single structure, or of any part of it which is free to vibrate. Under such circumstances of synchronism quite small and insignificant forces are capable of producing enormous results, as is well known to all engineers, and the most familiar illustration is perhaps that of a disciplined force crossing a bridge, who, if walking in step, are liable to set up dangerous oscillations in the most substantial of structures.

If the Blades of a Screw are of Different Pitch, especially at or near the tips, the pressure on opposite blades is not the same, the propeller will be running out of balance, and as each blade should take its due proportion of thrust to run in perfect balance, such differences in pitch or blade surface will easily cause vibration to be set up as the true centre of pressure will not coincide with the shaft axis, but have an orbit of its own, the centre of which is on that axis. The desire for the utter absence of vibration and the economy of high efficiency have induced some engineers to have the blades of propellers chipped and ground, so that, not only are the working faces smooth and true to pitch throughout, but each blade is of equal pitch with the others. In spite of the time and cost of carrying out this refinement, there is ample justification when the revolution is high and the ship a fast express steamer, or for good gunnery. For a cargo steamer it is, of course, quite desirable that the blades shall be of equal pitch, and, as far as possible, by care in moulding, to have a uniformity, but there is not the same warrant for chipping and grinding in their case. The same remarks apply to screws that are damaged by bending or breaking of the tips. It is, however, astonishing how badly damaged a screw may be and yet do its work fairly well.

The Screw is always working in Water disturbed more or less by the passage of the ship herself, as also by the currents set in motion by the suck of the screw. The former, known as wake currents, cause a difference of pressure on the upper and lower blades of the screw; the result is, therefore, pretty much the same as in the case of a screw with a blade out of pitch, except that in this case the real centre of pressure has no orbit but remains constant outside the shaft axis. The upper or surface current in wake of a ship has more forward motion than lower ones, and moreover is often quite a distinct layer of water flowing over practically still water, each blade in turn comes into it suddenly, and receives a sudden accession of load and a tendency to retardation of motion; if the period of blade stroke on this stream coincides or synchronises with the periodicity of the ship, it will soon set up quite sensible vibrations, and, if there is no break in the timing, they may become quite violent. If the synchronism is not perfect they will damp out and disappear, only to reappear later with equal violence. This kind of disturbance causes horizontal vibrations, which were very pronounced in ships having only two blades to a single screw; it was, and is, still noticeable with single-screw ships with four-bladed screws, especially when the tips are broad.

The Proximity of Screws to Portions of the Hull is also a common cause of vibrations, especially of the small or residual ones, that are often as troublesome as the larger ones. This is due to more than one agency. If the blade tip passes closely to a fixed obstruction, such as the stern frame, or the hull of the ship in the case of multiple screws, its resistance or thrust pressure is for the moment changed, and as each blade in turn does so, there is produced a series of impacts, which in themselves may be slight and almost imperceptible, yet may in their collective action, when synchronising with some massive portion of the ship, produce in it serious activities. There is always a film of water of quite sensible thickness dragged on by friction with the skin of the ship at a velocity not much inferior to the ship itself. If, therefore, the blade tip is so close as to come into that envelope of inert water from water that was flowing past the screw at the rate of 2,000 to 3,000 feet per minute, it would be almost as bad in its effect as if it struck a solid, and the blow each time a blade tip passed through would be, and probably is, very considerable, and the result is manifest on Dr. Schlick's interesting pallograph diagrams.

Then such obstructions as the stern frame and brackets, spectacle pieces, etc., act as brakes to the spiral currents from the screw tips, so that as each blade passes near it there is the change in direction of flow, which, although very slight in its effect, may in the cumulative form be very real.

The Power of Sucking in Air, so beautifully and clearly demonstrated by Professor Flamm, and experience by each observer of the screw in everyday practice, may, and probably does, lead to a considerable amount of vibration. It is well known that when air is drawn into the screw race there is a diminution of thrust, and there may be, and often is, a momentary cessation of it; if this becomes periodic, and the effects cumulative, a most unpleasant form of hull disturbance will be experienced. This air suction occurs the more readily with screws whose blade tips are near the surface, and even when there is a fair amount of immersion in still water the suction or feed to the propeller when running fast produces a hollow or depression of the surface, so as to render air "spouts" easy of formation. The same thing happens when there is quite a gentle swell on, so that the heave and pitch of the ship causes the blade tips to approach too close to the surface.

Cavitation may also be a cause of Vibration when the screws are quite thoroughly immersed, especially when the screw is driven by a steam engine with a torque of great variation, or with an internal combustion engine with the same or worse defect, so that the propeller has sudden and severe acceleration in angular velocity, whereby the pressure per square inch is increased beyond the limit of good working, so that cavities are formed by the reduction in pressure behind the blades, permitting of air separating from the water and forming masses of bubbles, which, when liberated, may have the same effect on the screw as the air drawn down from above by "suction."

All these Things may cause Vibration, and always are sources of loss of efficiency. The former can be generally reduced to a minimum, if not altogether damped out, by causing the engines to run at a rate of revolution that precludes synchronism with any of the important masses of the hull. The efficiency of propeller and hull can be little effected by such means, but doubtless the hull that does not vibrate is likely to permit of more gross power being applied to propulsion than one that does.

The Necessity for Perfect Balance of Engine and a quiet-running Screw is imposed on the designer, and he must study the questions involved as carefully as formerly he had to do those involving the safety and economic running of the machinery on shipboard.

The Auxiliary Machinery must also be free from Vibration and unevenness from the power to produce it; for to-day, with turbines and the beautifully balanced reciprocators, it would be grotesque to find the ship vibrating from a donkey pump's unbalanced forces; and now that air pumps, as well as all other pumps, are separable from the main engines, it is incumbent on their makers to supply them free from a vice, from which the main engines have been eradicated. The makers of electric-power generating engines have been compelled long ago to balance them so that they create no nuisance to the neighbourhood in which they are situated by themselves vibrating or causing any tremble in the neighbouring houses; therefore, as a rule, the makers of these engines who have had such shore experience can be trusted to supply them on shipboard quite free from vice.

The Introduction of Steel Castings was hailed with delight by engine designers thirty-five years ago, as it raised the hope that the weight of machinery generally and the cost of many of its parts would be very materially reduced, and that in other parts the risks run consequent on the necessary employment of cast iron, and even with cast bronze in their manufacture, would cease. To-day the engine designer is still living largely in the same hope, inasmuch as the sound, perfect casting at moderate cost in a metal strong and tough and capable of working at quite a high temperature remains yet to be produced in quantity and promptly. British steel founders can, and do, make huge castings, which for ship work are invaluable: they also supply engineers with some very fine and useful ones, but they had to look to another country for that soundness and uniformity in quality so much to be desired by those who are responsible for and take the risks of such things. It is a matter of great regret that there lacks sufficient certainty in the home product to encourage its more extended use for such parts of an engine as must be sound or machined all over. This is a disappointment to those who have looked year after year for the improvement so long hoped for.

The Use of Aluminium in engine construction has been somewhat delayed from somewhat similar causes, in spite of the fact that the material was sold at such a low price as to permit of its free use in engine construction. Now, however, the very beautiful castings supplied for motor car machinery must induce the marine engine designers to use them more freely in their products. The metal now used is an alloy, and is only a trifle heavier than pure aluminium; the castings are sharp and sound in outline and clean in surface; the tensile strength is sufficient, being as good as ordinary bronze. It will not, however, withstand the action of sea water as does bronze, so that it cannot be used for the parts exposed to it, but there are very many other parts which are quite free from contact with sea-water, and others where a coat of paint will give sufficient protection.

Duralumin, a 90 per cent. alloy of aluminium introduced by Vickers, Ltd., has such good qualities as soon to command the attention of the marine engine designer, inasmuch as it is quite strong (28 tons tensile with 15 per cent. extension), and will resist shock; it is very light (sp. gr., 2·8) and malleable, and can be treated as the zinc bronzes are, by drawing out into bars of various sections, also rolled into plates and sheets, and sold at quite moderate prices, so as to be extensively used in the construction of air-craft, motor cars, etc.; it cannot, however, be used for castings as yet.

In the Future the Reciprocator will probably undergo more changes, necessary to its success in competition with the steam turbine; and also that it may hold its own with the oil engine, it may be necessary to follow some of the features peculiar to that engine. For example, the oil engine at present has certain limitations, which hedge it in completely. The steam engine maker may impose on himself, if he chooses, some of them with advantage. For instance, the huge cylinders of the present triple and quadruple engine may give place to a multiplicity of smaller ones; so small may they be indeed that the low-pressure ones may have relatively such large ports and valves as will enable them to benefit usefully by the high vacua now attainable in condensers on board ship at a trifling extra cost.

Superheating of steam to an extent now practically impossible also may be followed with complete success in small cylinders, including a marked reduction in the consumption of fuel. There may be even modifications in the methods of reversing the engines, whereby there will be effected savings in first cost, as well as economy in working costs The flexibility of the reciprocating steam engine is a strong and marked feature in the eyes of the marine engineer, especially of him who has to work and be responsible for the machinery of a ship; it should count likewise with owners, underwriters, and "all those who go down to the sea in ships" as a fair set off for such savings in fuel as claimed, even though they be substantial; especially should this consideration weigh heavily in the counsels of citizens of a country having an abundant supply of excellent coal but little oil, and that little limited to one or two remote districts. To have to import food into a country is regrettable. but to import fuel where it can be avoided is folly. If our ships are to be largely fitted with internal combustion engines, or have boilers capable of burning oil fuel only, its state in case of a war with a country having a powerful fleet will be a perilous one, and lead to a worse disaster than a temporary shortness of food.

CHAPTER X.

THE CYLINDER AND ITS FITTINGS.

The Cylinder is the most Important Part of the reciprocating engine, for on it every other part is more or less dependent. The capacity of the engine for developing power—that is, for converting the energy of the steam into mechanical work—depends on its size and efficiency. The efficiency of the cylinder for this purpose will depend largely on the design of the various ports, passages, and valves, while its mechanical efficiency will be affected by design and the quality of the material and workmanship of the cylinder and its parts, inasmuch as the losses in this member of the machine are chiefly due to friction aggravated by thermal conditions not present in other portions of mechanism. It is, therefore, very highly necessary that the utmost attention is bestowed in the first place on all the problems involved in the determination of size, the general design of it, and all its working parts, and the greatest care in the manufacture, both in the workmanship and the selection of materials. Finally, in the working of the engine, judgment as well as care is necessary in fitting and adjusting them, and constant attention during service, that the highest efficiency may be obtained.

The Size of the Cylinders can be calculated on first principles in quite a simple way, but to determine the *best* size, or that most suitable for each particular case, requires something more; it requires special consideration of all the circumstances involved in the case. In a general way, it may be taken for granted that the greatest economy, both in prime cost and in working expenses, is to be attained, with the smallest capacity of cylinder in which the maximum horse-power guaranteed by the builder can be developed. There are, of course, exceptions to this as to other equally useful practical rules, such exceptions must, of course, receive exceptional treatment.

The marine engine differs in many important respects from the land engine of similar design; in one respect the conditions of service are very dissimilar, for whereas the land engine usually runs at a fixed rate of revolution when employed for continuous work, as in driving mills, dynamos, etc., the marine engine varies in revolution as the speed of the ship changes. Further, the load on a land engine may, and often does, vary from nothing to the maximum power with only a very slight, and that a temporary, variation in rate of revolution; whereas in the marine engine the gross horse-power developed varies roughly as the cube of the rate of revolution—that is, if a marine engine is slowed down by throttling or "linking up" to half the rate of revolution at full power, the horse-power will be only one-eighth of it. Even a reduction of 10 per cent. in speed of ship and revolution involves a reduction in power of no less than 27 per cent. Now, as the marine engine, even in express steamers, is seldom worked at full power, that developed on service is generally well below the full capacity of the engine; the periods of full power development are few and of short duration, and of such a nature is the service that demands them, that the cost of producing the power is insignificant compared with that of the general working, so that if extravagant by comparison with the ordinary expenditure, it is, nevertheless, quite warranted. In other words, so long as the increase in power is obtainable on demand, the cost of it is of little consideration; consequently the efficiency under those conditions may be quite low, if by these temporary sacrifices the gain in efficiency under the normal conditions of service is thereby attained and substantial. High efficiency during the long periods of service is, therefore, the first consideration of the designer, as well as that of the purchaser of the engine, and just as electrical engineers on shore demand engines which can on an emergency, and for a short time, develop something like 20 per cent. more than their normal maximum output, and call it "overload," so the marine engineer should be content with such overloads on the same terms for trial trips and spurts at sea, instead of requiring a larger engine, which will really run the greatest part of its life at only 75 per cent. of the full power at which it can run, to do that full power at maximum efficiency.

To the Engines of Warships the same argument may be applied in a general way, but to them some other considerations are applicable which may modify the decision of the designer. Here, in order that the utmost power may be obtained under the more restricted conditions of weight and space, there may be need of the highest efficiency, in order to get the greatest possible output of power from the boilers. But, judging by modern practice in such ships, the small cylinders, with the necessary low rate of expansion for them at full power, prevails to a greater extent than in even the mercantile marine, because a warship seldom runs at even so small a reduction of speed as the 10 per cent.; with her a reduction of speed of 40 per cent. is common, and that means only about 22 per cent. of the full power is required.

The Larger the Cylinders are the more costly they must be to manufacture and maintain; moreover, the clearance losses and those from condensation will be greater in the larger than the smaller cylinders. Further, the pipes and connections, the rods, valve gears, columns, shafts, and framing vary with the size of the cylinder, many of these parts being influenced by the maximum rather than the mean pressure or load on them. It is, of course, obvious that for a given power any decrease in cylinder capacity must be accompanied by a decrease in rate of expansion, all other things being the same. But all other things are not the same; the steam pressure from the boilers and the pressure in the condenser are the same, but the steam efficiency in the smaller set of cylinders may be such that the actual mean pressure at the same normal rate of expansion under working conditions is considerably greater than in the larger ones, due to less wire-drawing, radiation, etc., cylinder condensation, and decrease in back pressure due to larger ports, etc.

Economy is found, therefore, in Practice with rates of expansion lower than theory indicates, and consequently of late years great increases of boiler pressures have not been accompanied by the corresponding increases in rate of expansion such as was formerly anticipated. Hence, for the full powers required for trial speed the rate of expansion is expressed by $p \div 15$, where

p is the absolute pressure of the steam supplied to the engine. Following this rule—

$$\text{Rate of expansion of triple-stage engines steam } 180 \text{ lbs.} = \frac{195}{15} = 13.$$

$$\text{,, quadruple ,, } 210 \text{ lbs.} = \frac{225}{15} = 15.$$

$$\text{,, ,, ,, } 225 \text{ lbs.} = \frac{240}{15} = 16.$$

On service such ships will work with rates of expansion of 16 in the triples and 18 to 20 in quadruples.

In Naval Service and Express Cross-Channel Steamers the rates of expansion are somewhat less, so that their measure is made by dividing the absolute initial pressure by 18. For the triple-expansion engines of such ships with steam at 200 lbs. gauge, or 215 lbs. absolute—

$$\text{Rate of expansion} = \frac{215}{18} = 12.$$

The Service Speed of Short Distance Express Steamers is about 5 per cent. under the maximum or trial trip speed, and since the mean pressure referred to the L.P. cylinder will vary approximately with the square of the speed, the mean pressure on service will be $(0{\cdot}95)^2$, or 0·9 of that on trial. In designing the engine, therefore, it should be borne in mind that it is desirable to have maximum efficiency with a mean pressure 10 per cent. below that of the maximum possible from the cylinders, and as a corollary the cylinders must be large enough to develop 11 per cent. more power than when on service—that means the "overload" is 11 per cent. in pressure, but it will be 16·7 per cent. of the I.H.P.

Instead of Speed Margin with such Ships, it would be better to have a power one, for it is seen that with only 5 per cent. of the former there is 16·7 of the latter, and, after all, it is more important in a general way to have a reserve of power, which will ensure maintaining the service speed.

With Cargo Steamers there is a Disturbing Element not experienced with the warship or passenger steamer—viz., the difference in draft of water with different cargoes, and the possible difference in speed of revolution—in fact, it is equivalent in effect on the engines of a change in size of propeller. The engines can now move at a much higher rate of revolution with a smaller mean pressure of steam, and unless checked by throttling or "linking up," proceed at a rate far in excess of that anticipated by the designer, and provided for by him. Under such conditions the propeller is very liable to sudden and severe racing, in spite of governors and hand governing. Such engines must, therefore, for safety sake, have in themselves that which goes toward restraint under such conditions.

The Indicated Horse-power of an engine is the product of the mean pressure, the piston area in square inches and feet passed through by the piston in a minute divided by 33,000. It may be said, therefore, that the power depends on the area of the piston on the speed of piston and the pressure on it. With a given engine the two latter may be varied arbitrarily, inasmuch as the pressure may be increased or decreased by the throttle valve at will, or may be altered permanently by changing the cut-off of the distributing valves on the cylinders; the speed of piston, all other things remaining the same, being dependent on it will vary with the mean pressure. But with the same mean pressure the piston speed may be permanently reduced by increasing the pitch of the screw. When designing an engine the mean pressure and speed of piston intended must be decided on in order to calculate the area of piston. As a matter of fact, it is really the rate of revolution that must be decided on as a basis of calculation for a reciprocating engine, just as it is for a turbine. With a vertical engine under ordinary conditions the length of stroke may be generally of any reasonable amount, so that the real limit, after all, is the practical one of how fast may a piston move with safety and consistent with continuous good working.

It must not be overlooked that the *maximum velocity* of movement of a piston is $\frac{\pi}{2}$, or 1·571 times the mean, so that if the mean speed is 1,000 feet per minute, the velocity of the piston at about the middle of the stroke is at the rate of 1,571 feet.

The Diameter of Cylinders of a Marine Engine is ascertained by first determining that of the low pressure from the referred mean pressure and speed of piston decided on, and making the high pressure of such a size as to permit of that mean pressure with the boiler pressure provided. The generality of engines now in use are of the compound type, triple or quadruple. The rate of expansion will be chosen with regard to the conditions under which the engine has mostly to work. In a general way it is true, as already stated, that the smaller the cylinders are the better, both in prime cost and economic working. It has been shown that the consumption of steam in Naval reciprocating engines at full speed per I.H.P. per hour is somewhat higher than at a somewhat reduced one, while, on the other hand, at low speeds, notwithstanding the higher rate of expansion obtaining, the consumption is often even greater than at full speed. This means that with the comparatively small cylinders the economy at an extravagant rate of expansion is better than that in the same cylinder at a high and presumably economic rate.

In the mercantile marine the lowest rate of steam consumption per horsepower is also often at a higher rate of expansion than that at the maximum power, but not much more so. With the cargo steamer, where economy of fuel is of prime importance, and the engines are run nearly always at full speed, the consumption of steam per I.H.P. is a minimum, or nearly so at that speed. Mr. Parsons found that the steam consumption of the triple-expansion engines of the s.s. "Vespasian" at 70 revolutions was 16·9 lbs. per horse-power hour, at 65 revolutions it was 17·6 lbs., and from that the rate gradually rose as the revolutions fell, till at 50 revolutions it was as much as 19·8. At 65 revolutions the rise was small, as it was only 17·6 lbs., and, assuming the same rate of slip, the following comparative results are true. The total consumption of water at 70 revolutions was 17,250 lbs. per hour, while at 65 revolutions it was 14,200 lbs. For the same *distance*, therefore, if at 70 revolutions the consumption is 17,250, while when done at 65 revolutions it will be only 15,140—that is, as 1·14 to 1·0—or a saving of 14 per cent. in spite of an increase per I.H.P. of 4·14 per cent.

RESULTS OF TRIALS OF VARIOUS SHIPS, SHOWING CONSUMPTION OF STEAM AND FUEL IN LBS. PER H.P.-HOUR.

Max. I.H.P. Engines	H.M.S. Hn., 18,500 I.H.P. Recipros.		H.M.S. Cn., 23,500 I.H.P. Recip. Trpls.		H.M.S. Tz., 9,800 I.H.P. Recip. Trpls.		U.S.A. Dlw., 28,500 I.H.P. Recip. Trpls.		H.M.S. As., 24,000 I.H.P. Recip. Trpls.		H.M.S. Hfr 10,400 I.H.P. Recip. Trpls.		I.J.N. Hk., 27,000 S.H.P. Curtis Turbs.	
	Steam.	Coal.	Steam.	Coal.	Steam.	Coal.	Steam.	Coal.	Steam.	Coal.	Steam.	Coal.	Steam	Oil.
Full power,	17·2	1·80	16·0	1·99	13·91	2·65	*21·0	..	*19·9	2·03	..	1·407	13·77	1·10
50 to 75 %,	15·2	1·76	14·4	2·04	15·45	2·28	18·7	...	15·37	1·85	..	1·49	14·76	1·27
20 to 10 %,	15·1	1·94	15·3	2·00	16·25	2·06	22·0	..	16·95	1·88	..	1·64	20·57	1·80

* This consumption was for all purposes.

Max. S.H.P., Engines,	R.M.S. La., 64,600 S.H.P. Direct Turbs.		U.S.A. Nvd. 23,300 S.H.P. Geared Turbs.		U.S.A. Fda. 40,500 S.H.P. Direct Turbs.		U.S.A. Pkn., 11,700 S.H.P. Curtis Turbs.		U.S.A. Esn., 17,800 S.H.P. Geared Turbs.		R.M.S. Ok., 6,900 S.H.P. Mixed R. & T.		U.S.A. Clfa. 29,000 S.H.P. Electric Turb.	
	Steam.	Coal.	Steam.	Oil.	Steam.	Coal.	Steam.	Oil.	Steam.	Oil.	Steam.	Coal.	Steam.	Oil.
Full power,	12·77	1·43	15·40	1·32	13·40	1·59	14·49	1·42	15·76	1·23	14·12	..	11·90	..
50 to 75 %,	13·90	1·56	16·16	1·30	..	..	15·50	1·40	17·40	1·22	14·10	..	11·10	..
10 to 20 %,	21·23	2·52	22·95	1·72	23·77	2·23	21·49	1·67	28·75	2·03	15·20	..	14·60	..

It is to be observed that, whereas a marine engine slows down as the rate of expansion increases, a similar engine on shore driving a mill or dynamo will run at constant speed. This is important, because economy of steam consumption is very much influenced by the rate of flow through the cylinders, just as it is in the turbine, the low consumption of which is in no small measure due to the rapidity with which the steam passes from entrance to the condenser. In a reciprocating engine, when at 150 revolutions, and of the triple-expansion type, the steam takes 0·8 second to pass through the system, but at 75 revolutions the time is 1·6 seconds.

Mean pressure is modified in quite a satisfactory way by linking up; for example, the s.s. "Al." was intended to work at sea with about 1,000 H.P.; on trial trip at full speed 1,232 I.H.P. was developed. The following shows how this reduction was effected by simply notching up to the extent of two turns of the hand wheel, and what were the consequences :—

TABLE XXX.—TRIALS OF A TYPICAL CARGO STEAMER.

	Steam.	Vacuum.	Revs.	Mean Pressures.			Indicated Horse-Power.			Total.
				High.	Medium.	Low.	High.	Medium.	Low.	
Trial Runs, full,	145	25·0	64·2	50·4	24·51	7·91	385·7	434·5	412·1	1,232·3
Sea speed, linked,	145	25·5	58·1	45·2	22·9	7·20	313·4	367·2	339·9	1,020·5

The coal was weighed during these trials, and the consumption was found to be at the rate of 1·51 lbs. per horse-power at full, and 1·55 lbs. at the reduced speed. Now, had smaller cylinders been provided, and the revolutions raised from 58 to 64, the rate of consumption would not have exceeded the 1·51 lbs., but probably have been less.

The Back Pressure in the L.P. Cylinder should be not more than 1·5 lbs. greater than that in the condenser—that is, with a vacuum of 28 inches it should not exceed 2·5 lbs. absolute. In a well-designed low-pressure cylinder at moderate piston speeds the vacuum in it at exhaust should be 95 per cent. of that in condenser, when that is not more than 28 inches; 93 per cent. is common experience with good engines when working with a vacuum of 27·0 inches at full speed. For purposes of calculation, therefore, an allowance of 90 per cent. will be quite on the safe side. With modern air pumps and a good condenser 28 inches can be maintained even in the tropics; 90 per cent. of this is 25·2—that is, the back pressure is about $2\frac{1}{2}$ lbs. If the ship is likely to see much service in the tropics, 27 inches will be safe, and 90 per cent. of it is 24·3 inches, so that the back pressure then will be 2·85 lbs. With a turbine the back pressure in it is very nearly that in the condenser top, as there is practically no exhaust pipe, and certainly no valve obstructions. For general traders the back pressure should be assumed at 3 lbs., and for those other ships casually visiting the tropics $2\frac{1}{2}$ lbs. is sufficient allowance for calculations.

Example (1).—To ascertain the probable mean pressure referred to the L.P. cylinder of a passenger steamer whose service will be through the tropics, the boiler pressure to be 185 lbs.

Here the initial absolute pressure is 200 lbs.

The rate of expansion $200 \div 15 = 13{\cdot}33$.

Referring to Table xvii. for this rate of expansion, the ratio $\frac{p_m}{p_1} = 0{\cdot}269$. Back pressure is 3.

Then theoretical mean pressure $= (200 \times 0{\cdot}269 - 3) = 50{\cdot}8$ lbs.

The actual mean pressure will be found by multiplying this amount by the factor given for such engines on (Table xviii., p. 125), say, 0·7. Thus:—

$$\text{Probable mean pressure} = 50{\cdot}8 \times 0{\cdot}7, \text{ or } 35{\cdot}56 \text{ lbs.}$$

The Diameter of the L.P. Cylinder may be found* then as follows:—When p_m is the referred mean pressure. Let d be the diameter in inches, R the number of revolutions, and S the stroke in feet.

$$\text{Then area of piston} = \pi\frac{d^2}{4} = \frac{\text{I.H.P.} \times 33{,}000}{p_m \times \text{R} \times 2\text{S}}.$$

$$\textit{Rule.}\text{—Diameter L.P. cylinder} = \sqrt{\frac{\text{I.H.P.} \times 21{,}000}{p_m \times \text{S} \times \text{R}}} = \sqrt{\frac{\text{I.H.P.} \times 42{,}000}{p_m \times \text{piston speed}}}.$$

If the stroke is not decided, S × R is half the mean piston speed.

The volume of steam at cut-off for a rate of expansion E must be = capacity of L.P. cylinder ÷ E.

The equivalent cut-off in the H.P. cylinder for this will be expressed as a fraction of the stroke = ratio of cylinders ÷ E.

If the cut-off is to be early, as it should be, to avoid excessive "drop,"

* The N.E. Coast Inst. E. and S. Rule for this in cargo ships is:—

$$D = \sqrt{\frac{\text{I.H.P.} \times 21{\cdot}9}{\text{S} + 4}}; \text{ the I.H.P. is that developed while on voyage.}$$

it will be seen that the ratio of cylinders must be smaller than if a later cut-off were admissible or desirable.

Then, if x be the cut off as a fraction of the stroke, and clearance is neglected

$$\text{Then the ratio of L.P. to H.P. cylinder} = \text{E} \times x,$$
$$\text{or } x = \text{cylinder ratio} \div \text{E}.$$

Taking the cut-off at 0·55 of the stroke, the ratio of L.P. to H.P. will be 0·55 × rate of expansion, and if that is taken as absolute pressure ÷ 13 for economic engines.

Rule.—Ratio of L.P. to H.P. cylinders $= \dfrac{0{\cdot}55 \times \text{absolute working pressure}}{13}$, or $= \dfrac{\text{absolute working pressure}}{23{\cdot}6}$.

Example.—To find the sizes of cylinders for a ship whose engines are to develop 3,600 I.H.P. on the conditions named before, with the piston speed at a mean of 700 feet per minute.

Here the diameter of cylinder . . $= \sqrt{\dfrac{3{,}600 \times 42{,}000}{35{\cdot}56 \times 700}} = 78$ inches.

Ratio of L.P. to H.P. cylinder . $= \dfrac{0{\cdot}55 \times 200}{15} = 7{\cdot}3$.

Diameter of H.P. cylinder . . $= \sqrt{\dfrac{78^2}{7{\cdot}3}} = 28{\cdot}8$.

The Size of Intermediate Medium-Pressure Cylinder to avoid drop should be larger than it would be if determined by such practical considerations as low initial loads to avoid shock, variation in temperature, etc. The true mean between the L.P. and H.P. cylinder would be as follows:—Where d is the diameter of the H.P., and D that of the L.P. cylinder.

$$\text{Diameter of M.P. cylinder} = \sqrt{\frac{d^2 + D^2}{2}}.$$

This, however, is too large, and not in accordance with practice; the following rule may be followed, therefore, with triples:—

Rule.—Diameter of medium-pressure cylinder $= \sqrt{\dfrac{d^2 + D^2}{3}}$,

or, Simply ,, ,, $= (d + D) \times 0{\cdot}45$.

For the engine in the above Example,

$$\text{Diameter medium cylinder} = \sqrt{\frac{28{\cdot}8^2 + 78^2}{3}} = 48{\cdot}1.$$

For such diameters of cylinder in an ordinary passenger steamer the stroke would be 48 inches. Then the engines would have cylinders $28\frac{3}{4}$, 48, and 78 inches diameter, and 4 feet stroke, the revolutions are then 700 ÷ 8, or 87·5 per minute.

The Arrangement of the Cylinders of a marine engine is now a much more important, as also a more interesting, problem than was the case formerly, when there were only two of them. To-day, with the multiplicity of cylinders and cranks that are commonly found, especially in oil internal combustion engines, the arrangement is governed by circumstances quite beyond the ken of the older engine builder, who was never troubled with problems involving the nice balancing of the moving parts so as to avoid vibration, or to study the economy of steam piping involved in determining the sequence of cylinders so as to get the best flow of steam from boiler to condenser with the least expenditure on balance weights.

Now the designer has to face such questions, as well as all that is involved in the division of cylinders to bring them within reasonable limits of size when the power to be developed is large. Large cylinders can be made as well now as formerly, but modern engineers prefer to have two of moderate size in the place of the one of 130 or even 140 inches diameter fitted by their predecessors, even though the highest pressure in them is only half that in these older and larger ones. Moreover, the solid steel pistons of to-day are lighter and certainly safer than the old hollow cast-iron ones, and in the vertical engines they are so very much easier to examine, overhaul, or remove than those of the horizontal engine of old days. Nevertheless, the tendency is to split the L.P. member of a compound system into two or more portions, for thereby a triple-stage engine may have four cranks, and be balanced on the Schlick system, and each L.P. cylinder can have a larger ratio of port and steam passage section, whereby the back pressure in it is less than that obtaining in the single large cylinder, and thereby benefit by a high vacuum in the condenser.

The Size of the Steam Ports in a Cylinder do not vary in practice with the *square* of the diameter, but at a somewhat less rate, inasmuch as the transverse measurement may be in proportion to the diameter, while the longitudinal or axial breadth does not increase so rapidly, therefore the larger is the cylinder the smaller is the ratio of maximum port to piston area; consequently with the same speed of piston the flow of steam is of necessity higher through ports and passages. If, therefore, the number of cylinders be multiplied, so as to keep their sizes comparatively smaller, the higher will be the steam efficiency. The mechanical efficiency, however, will probably be considerably less, on account of the multiplicity of working parts, glands, etc. On the whole, however, within reasonable limits and with modern workmanship and material, the general efficiency of the engine with the larger number of cylinders is higher than any older one with the less.

There are other influences at work to-day, which incline the marine engine builder to more subdivision of cylinders. The oil engine, with its very high initial pressures and temperatures, requires not only a very strong and simple cylinder, but one of quite a limited diameter. Since the single-acting two-cycle engine is likely to be the type for competition with the turbine and reciprocator on shipboard, it is more than ever sure to have cylinders of limited diameter. The Diesel oil engine (fig. 49), using heavy oil of sorts, and without special ignitors, is looked to as the one specially suitable among internal combustion engines for marine purposes. This engine, with its initial pressure of 550 to 650 lbs. per square inch, and liable

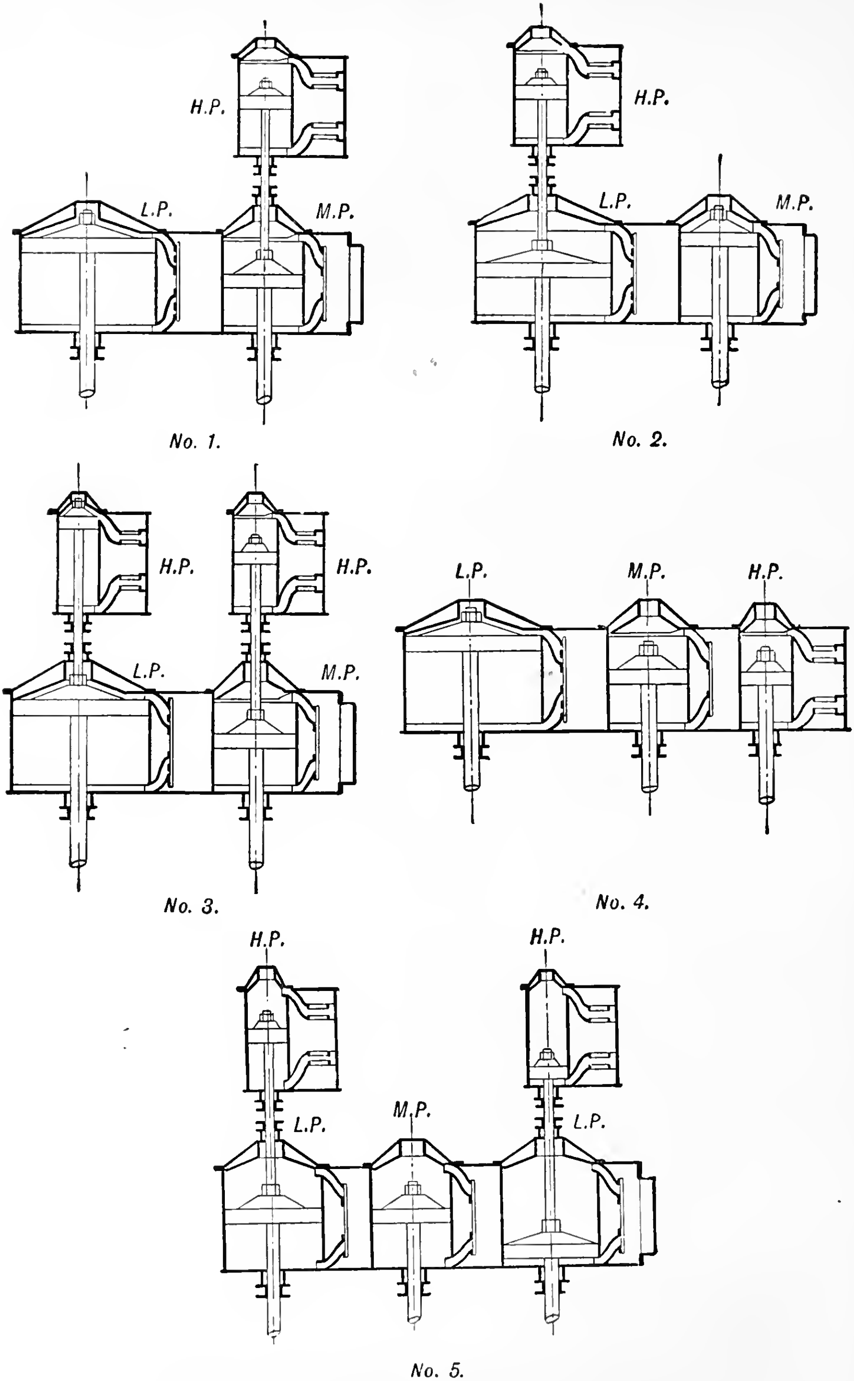


Fig. 73.—Various Arrangements of the Cylinders of Triple-Expansion Engines.

to a temperature exceedingly high, requires to be quite moderate in size and strong in construction.

Even the steam engine of to-day is subject to quite high pressures, and when the steam is superheated it may have a temperature of 600° F., with a pressure of 230 lbs. per square inch, which requires similar care and limitations as the oil engine, although in a lesser degree, however, to be observed in the design of the H.P. cylinder. Under these circumstances lubrication of the internal working parts is at all times uncertain, and never to be depended on as being positive, hence the smaller the cylinder, the less liability to serious derangement.

The Arrangements of Cylinders possible for a Triple- and Quadruple-Stage Engine may be studied by referring to their diagrams on figs. 73 to 74. The early triple-expansion engine of Dr. Kirk had always three cranks, with the natural sequence of cylinders (*v.* fig. 76). The very earliest triple-stage engine, however, had only two cranks, with the H.P. and M.P. cylinders in tandem over one and the low pressure over the other, as in fig. 73. No. 1.

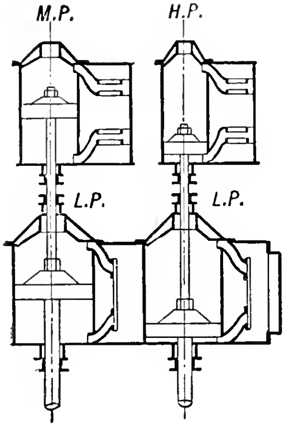
Inasmuch as the older compound engines were made with the L.P. rods and valve gear much heavier than those of the high pressure, it was found better, when tripling such engines, to fit the third or new H.P. cylinder over the low pressure (fig. 73, No. 2); the engine so treated was not only safer but better balanced and easier to start than when the new H.P. was placed over the original H.P. cylinder.

When the larger two-stage compound engines were tripled two new H.P. cylinders were fitted one over each of the old ones, as shown in No. 3, then the load was evenly divided on the cranks and the combination very quick in handling. Some of the old and large two-stagers were tripled by the addition of a complete new H.P. engine before the old ones, and coupled to them in the best way circumstances permitted.

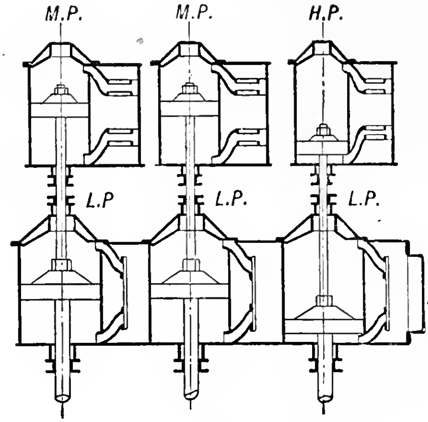
The old three-crank two-stage compound engines were tripled by adding a new H.P. cylinder tandem to each low pressure, and some new engines of large size have been made on this plan.

No. 1 design of fig. 73*a* is one rather for treating an existing expansive engine having two equal-sized cylinders than to copy for a new engine. If, however, large power were wanted in small floor space, but with unlimited height, it is quite a suitable arrangement. The same may be said for No. 2 on the same page when larger engines still are wanted on a limited floor space. In this case, however, there is a simplicity in design and limitation in dimensions that are attractive. In fig. 73*a*, Ex. 2, there are three cylinders in line of equal diameter; above them tandem-wise are three other cylinders in line and of equal diameter. Of these the first is the high pressure of the system, the next two are the medium ones, and the lower three are the L.P. cylinders. The ratio of M.P. to H.P. is thus 2, and as that of the L.P. to the H.P. is 6, that of L.P. to M.P. is 3. As a concrete example, the L.P. cylinders may be taken as each 60 inches diameter, the H.P. and two M.P. cylinders are therefore $= \sqrt{\frac{1}{6} \times 3 \times 60^2} = 42{\cdot}4$ inches diameter.

Diagram 3 (fig. 73*a*) shows the four-crank three-stage compound arrangement of cylinders, with the two low pressure of equal weight of moving parts at the ends, as is the common practice. Diagram 4 shows a variation, with the two L.P. cylinders next one another, with a valve box common to the two and their cranks at right angles, so that the flow of steam is



No. 1.



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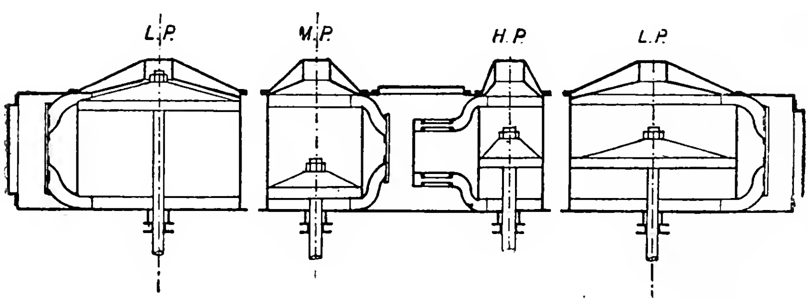
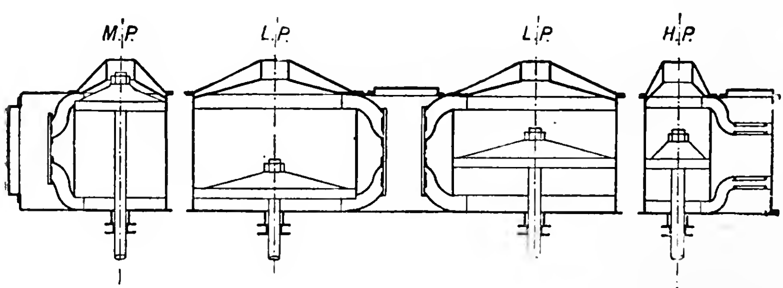


No. 3.

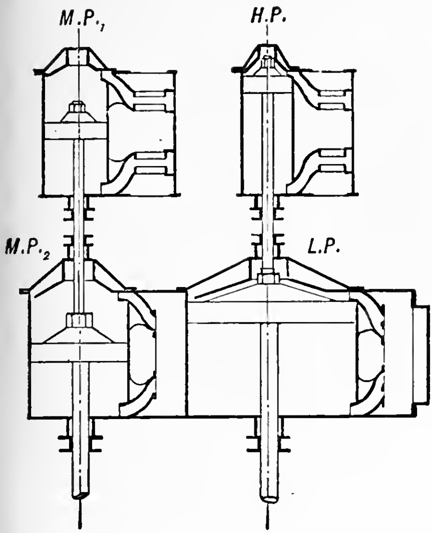
Four cranks, L.P. rods reduced.



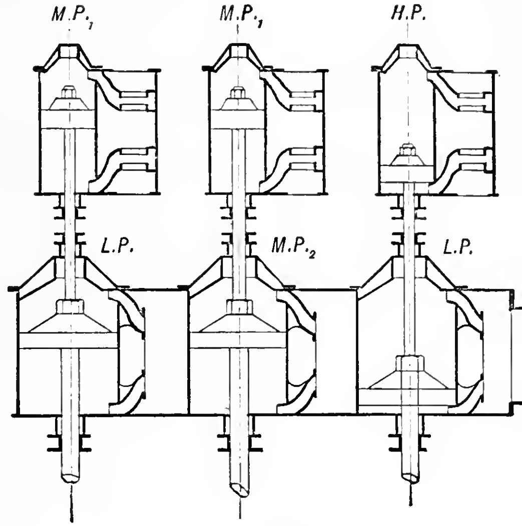
No. 4.

Four cranks, all rods same size.

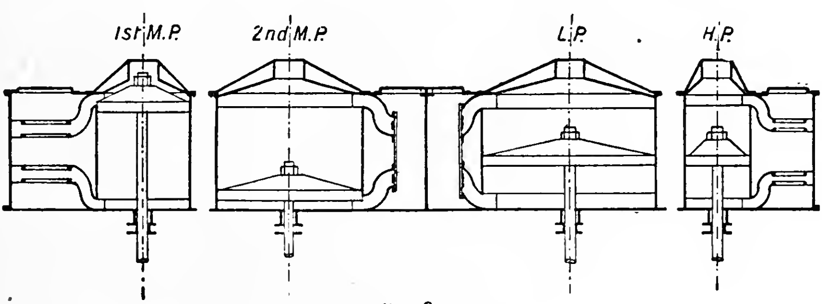
Fig. 73a.—Various Arrangements of the Cylinders of Triple-Expansion Engines.



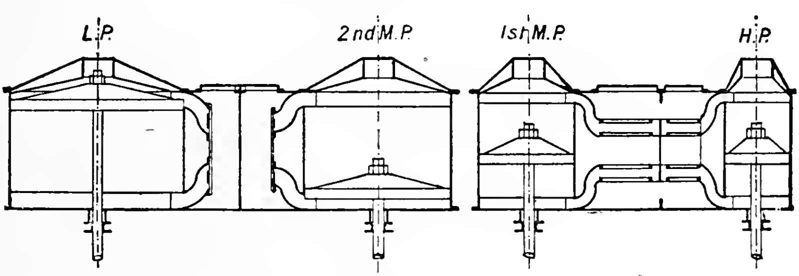
No. 1.



No 2.



No. 3.
Four Sets of Valve-Gear.



No. 4
Two Sets of Valve-Gear.

Fig. 74.—Various Arrangements of the Cylinders of Quadruple-Expansion Two-, Three-, and Four-Crank Engines.

practically continuous through quite short exhaust pipes, and the flat L.P. valves are fitted with a balancing frame between them.

The Arrangement of Cylinders of the Quadruple Engine were originally as shown on diagram 1, fig. 74 and fig. 75, in order that a compact engine taking little floor space should have an advantage over the three-crank triple. Moreover, the four-stage compound system was capable of a good arrangement of valves when in tandem, as may be seen in fig. 75. Another tandem arrangement is shown in diagram 2 suitable for large engines and three-cranks. As in the case of the triple-stage six-cylinder engine, there may be

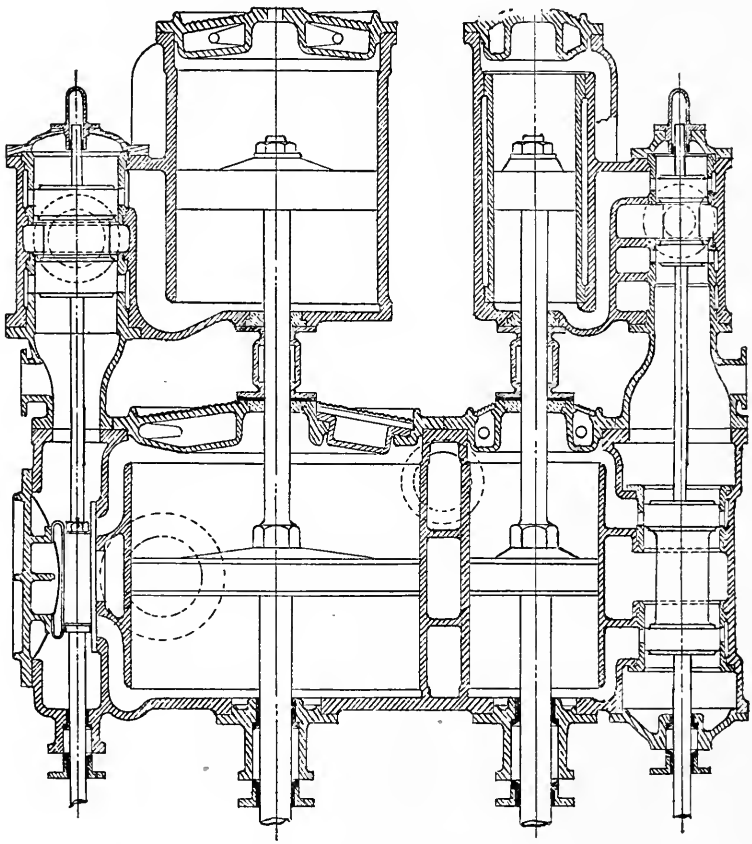


Fig. 75.—Cylinders of a Two-Crank Quadruple-Expansion Engine.

with the quadruple-stage only two sizes of cylinder, for here the first of the lower three may be the second M.P. cylinder, while the second two of the upper three are the first M.P. cylinders. Then, if the ratio of L.P. to H.P. is 8, then the upper ones will be just half the diameter of the lower; the ratio of second M.P. to first M.P. will be 2, and of L.P. to second M.P. cylinder also 2. Thus, if the lower cylinders are each 60 inches diameter, the upper ones will be 30 inches each; quite a simple arrangement.

Designs 3 and 4 show the cylinders as arranged for four cranks, the latter being the natural sequence of cylinder, while with the former the smaller cylinders are outside and the three last in sequence, so that the rocking action is reduced by the arm of the couple being a minimum.

Figs. 69 and 70 are examples of multi-cylinder engines operating on one crank, the axis of each engine being virtually in the same transverse plane as the others. Considerable fore and aft space is saved by such a design, and although very much out of balance statically, such engines will work satisfactorily, and are quite suitable and useful for tug boats and river craft working in smooth water, where space and prime cost are of great importance. They are largely employed on the American rivers and harbours, where the tonnage question does not affect the designer.

Fig. 283 is another example of compactness, whereby four cylinders are caused to operate on two cranks without being in tandem. In this case, however, the engines are balanced both statically and against inertia forces.

The Ratio of Cylinders in Practice depends somewhat on the service of the ship. The cargo boat, with triple three-crank engines and a working pressure of 185 lbs. the ratio of L.P. to H.P. cylinder is about 6·5,* while with a higher boiler pressure up to 200 lbs. it is 7·0. With express steamers and such pressures the ratios will be from 5·5 to 6·0; in Naval ships the ratio is no more than 6·3 when the working pressure is over 200 lbs., and 7·0 when 250 lbs. Quadruple engines are limited to service in the mercantile marine and with them the cylinder ratios are from 8 to 9, the latter being the rule with boiler pressures up to 230 lbs. per square inch. In small high-speed craft, as torpedo boats, destroyers, and other similar ships, the ratio with 180 to 200 was only 4·5 to 5·0.

To determine the Ratio of the L.P. to the H.P. cylinder in any compound system, the following simple rule holds good:—When p is the absolute pressure of the steam at H.P. valve-box, ratio of cylinders $= p \div K$, where in the mercantile marine K is 27 for cargo steamers, and 34 for express steamers. In the naval service K is 35 for cruisers and battleships, and 42 for scouts and destroyers. The economy of these vessels at low speeds is quite good, and even at high are not so very extravagant considering the low steam efficiency.

Having determined the Diameter of the L.P. Cylinder necessary for the power required from the engine, and deduced from the conditions and circumstances of the particular case the diameter of the H.P. and M.P. cylinders, the designer can proceed with the dimensions and arrangements of all that pertains to them. It is, however, of prime importance that the pipe through which the steam is supplied to the engine from the generators, and that through which it passes away from it to the condenser are of adequate size, and as short as circumstances will permit, for otherwise there may be a drop of pressure more than is desirable from the boiler to the valve chest, and, what is worse still, a more serious drop from the L.P. cylinder to the condenser. If the ratio of L.P. to H.P. cylinder is 7·0, the loss of a pound of pressure at the L.P. cylinder will require an increase of 7 lbs. mean pressure at the H.P. cylinder to make compensation. Further, if the referred mean pressure can be raised from 33 to 34 lbs. by a good condenser and adequate exhaust pipe, the gain in mean pressure is 3 per cent., and the gain in power somewhat more if the engine is quickened by it as it would be. In all reciprocating engines the

* N.E. Coast Inst. E. and S. have adapted 7·5 in their standard specification for 180 lbs.

flow of steam to and from them is intermittent more or less; the higher the rate of revolution the smaller is the variation due to it. With slow-running

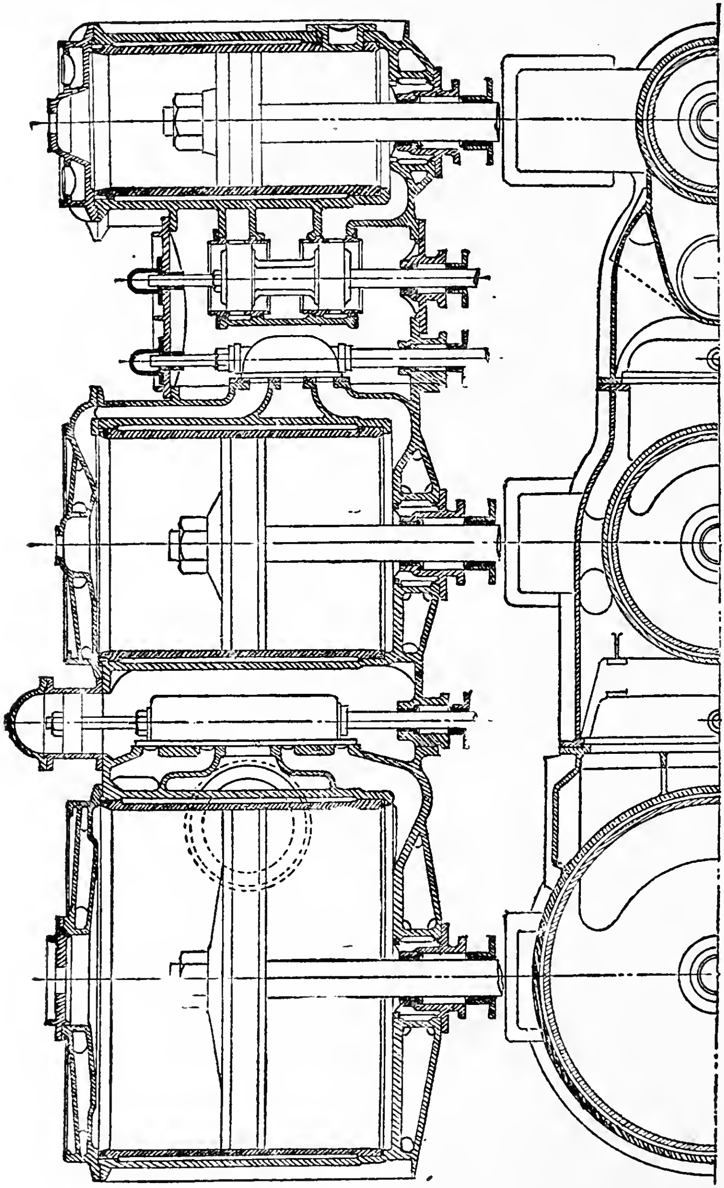


Fig. 76.—Cylinders of a Three-Crank Triple-Expansion Engine.

engines the H.P. valve-box may be with advantage of such a size as to act as a receiver, and so prevent the wire-drawing action at entry of steam being excessive; but there can never be the steady flow that is possible with a turbine, whereby there is no check to the stream with its consequent pulsation that must detract from the efficiency of the reciprocator. Then there must be likewise adequate means for conveying the steam from cylinder to cylinder with least loss in the process. There must be, of course, some drop in pressure at every stage, as without it there could be no flow of steam, but the drop should be small, and only a means to an end, and not a cause of loss beyond the actual needs. If the flow is practically continuous, the sizes of pipes and passages may be quite moderate compared with what they must be in a slow-running engine with intermittent demand. This must be remembered, that at the top of the stroke the acceleration of piston velocity is more rapid than at or near the bottom, and that with an early cut-off the velocity is less than the mean, so that calculations based on mean piston velocity will give ample areas of cut-off, etc.

Since with slide valves at both steam entry and exhaust the orifice is an expanding and contracting one, while the section of the channels or passages is constant, and considering that clearance space is detrimental to economy, the latter need not be of the same area as the ports; or, what perhaps is the safer *dictum*, the ports should always have a larger area than the passage sections (*v.* fig. 77). Also the passages should be made as short as possible, and as free from turns and corners where eddies may set up as the general design permits; especially should this be followed with the L.P. cylinder where the loss from clearance is irrecoverable.

Drop Valves have been used successfully with engines of the marine type on shore; these, however, are not required to reverse, and do not as a rule work a 24-hour day or a 7-day week; moreover, a stoppage from any cause is not likely to be fatal at any time. If such valves could be trusted on board ship, and arranged for reversal, much of the steam and thermal losses of the marine engine would be avoided, and besides, the pretty indicator diagrams produced be without the drawbacks, for which, from the marine engineer's point of view, they are inadequate compensation. Judging by what is occurring in the motor-car world, the drop valve is not the desideratum of the oil engine builder, for one by one they are reverting to slide valves.

Main Steam Pipe.—The main steam pipe, which supplies a cylinder with steam, should be of such a size that the mean velocity of flow through it does not exceed 8,000 feet per minute. When this is not exceeded, the loss of pressure between the boiler and the valve-chest is very slight indeed. If, however, the valve-chest is large, the cut-off in the cylinder is before half-stroke and the rate of revolution moderate, the area of transverse section of this pipe may be smaller than given by this rule, for as stated the piston speed is below the mean velocity at the early part of the stroke, and the space in the steam-chest acts as a reservoir for steam, so as to keep up a steady supply during admission. If the space is not less than one-half the volume swept through by the piston at cut-off, the velocity of steam in the pipe may be *assumed* to be 9,000 for engines of 150 N.H.P. to 250 N.H.P., and 10,000 for those above that power; for smaller engines, owing to the comparatively larger resistances of small pipes, it is not advisable to take a higher speed than 8,000. On the other hand, if the cut-off is later than half stroke, and the valve-box small, the assumed velocity should be at least 10 per cent. less than that given above.

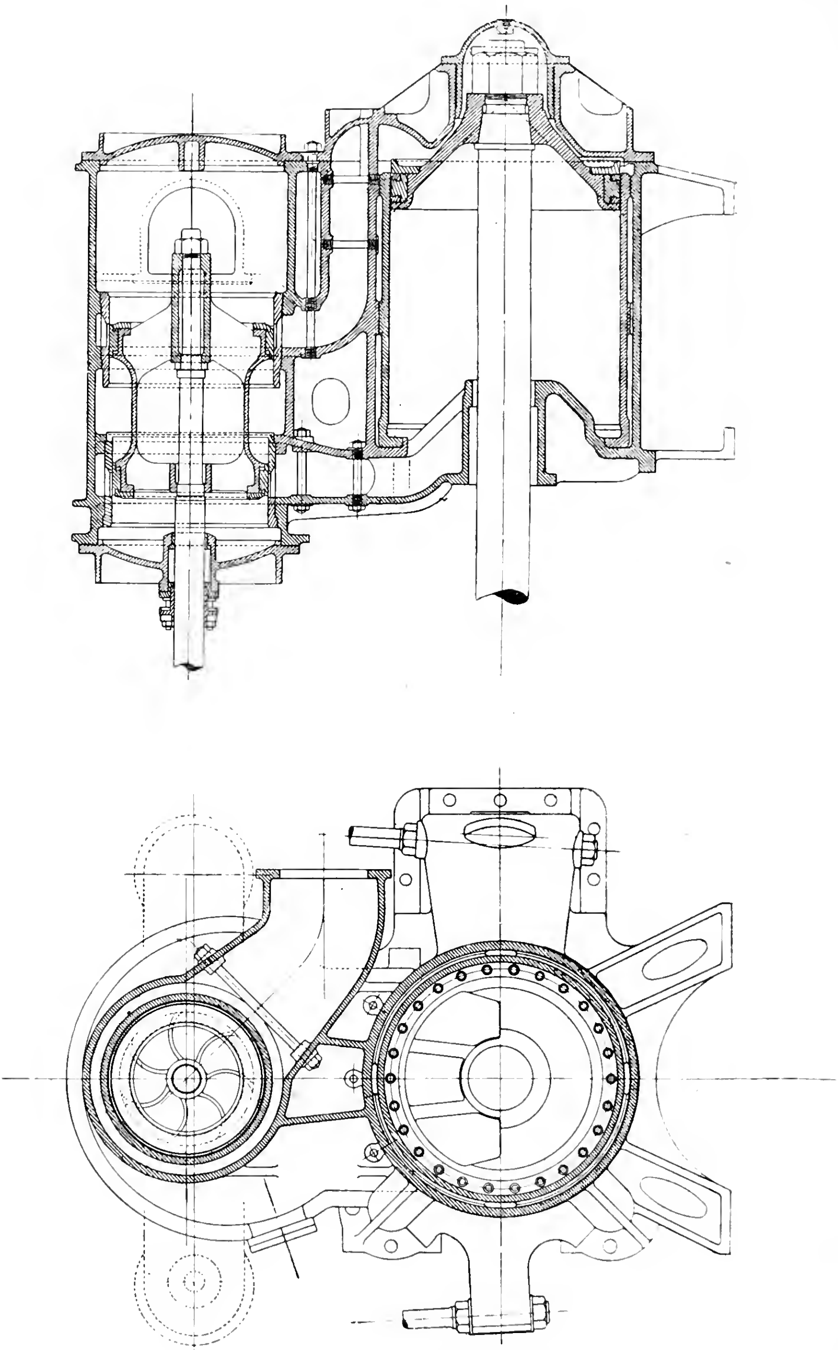


Fig. 77.—H.P. Cylinder (Naval).

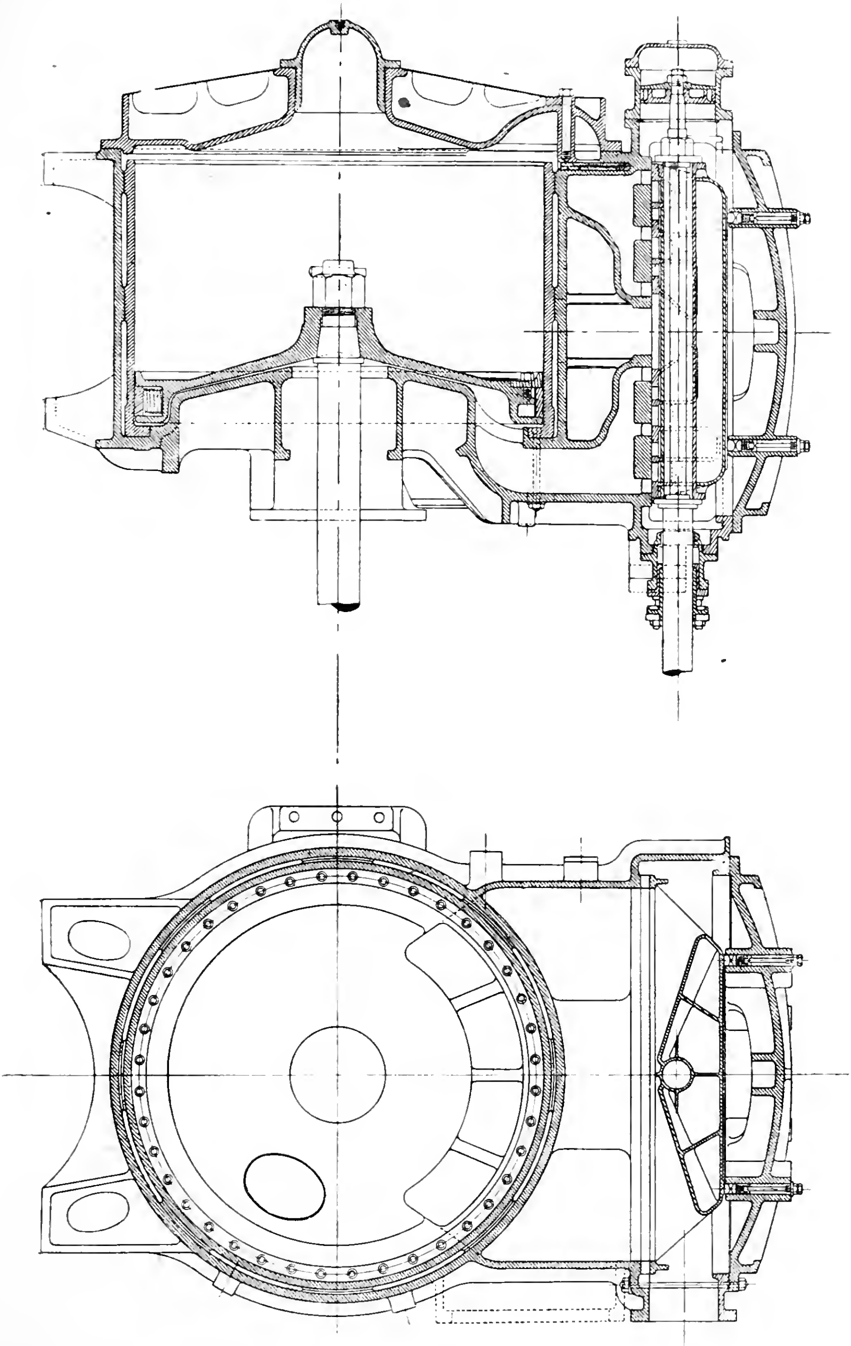


Fig. 77a.—L.P. Cylinder (Naval) with Triple Steam Parts.

Taking 8,100 feet as the mean velocity, S the mean speed of piston in feet per minute, and D the diameter of the cylinder, then,

$$\text{Diameter of main steam pipe} = \sqrt{\frac{D^2 \times S}{8,100}} = \frac{D}{90}\sqrt{S}.$$

Example.—To find the diameter of the main steam pipe to a cylinder 45 inches diameter and 5 feet stroke, the revolutions at full speed to be 60 per minute.

Here $S = 2 \times 5 \times 60 = 600$, and $D = 45$ inches. Therefore,

$$\text{Diameter of main steam pipe} = \frac{45}{90}\sqrt{600} = 12{\cdot}25 \text{ ins.}$$

When the main steam pipe is abnormally long, as is the case with large ships with long boiler rooms, the divisor should be 85 to 87.

Area through Stop and Throttle Valves.—Although the loss of pressure at the valve-box is often attributed to want of sectional area in the main steam pipe, it is more frequently due to contracted area past these valves. The friction through a number of small openings is considerably more than through one of an area equal to the collective areas of those openings, especially if the sum of the perimeters of the latter largely exceed that of the single opening, and the "loss of head" will be large if due allowance is not made. For this reason there should be always an excess of area around valves and other obstructions to the free passage of steam, and the passages leading to and from them should be as easy as possible, so as to avoid violent changes both of direction and velocity of flow.

Steam Ports and Passages.—Since, in most engines, the steam has to exhaust through the same ports and passages by which it was admitted, their size must be governed by the proper flow of emission rather than of admission. The area of section of steam ports should be such that the mean velocity of flow at exhaust should not exceed 6,000 feet per minute. The ports should be somewhat larger than the section of the passages, especially when certain kinds of valves are used, which will be dealt with later on; as a rule, they have nearly the same area as the sectional area of the passages. To avoid excessive *clearance*, however, the capacity of the passages should be as small as possible consistent with free flow of steam, and as this depends greatly on their sectional area, so that the reduction in capacity can only be attained by making them as short as possible. Not only will the evils arising from *clearance* be avoided, but the loss through resistance be very materially lessened by shortening the distance between the valve face and cylinder, and their cooling effect on the incoming steam temperature.

Area of steam ports and of section through passages

$$= \frac{\text{Area of piston} \times \text{speed of piston}}{6,000}$$

$$= \frac{(\text{Diameter of cylinder})^2 \times \text{speed of piston}}{7,636}.$$

Opening of Port to Steam.—It is advisable so to design the valve, etc., that the opening for admission of steam to the cylinder is sufficient to avoid any serious loss by "wire-drawing;" but in actual practice, unless special gearing is designed so as to give a quick motion to the valve at the instant of cut-off, there is very considerable loss of pressure shown on the indicator-diagram; and, what is worse still, from deficient opening, the loss is generally not limited to the period near to cut-off, but during the whole time of admission. The ordinary valve gears do not give that quick motion, either at opening or at cut-off, which is such a desideratum. Separate expansion valves and special valve-gearings admit of such a motion, and consequently the opening to steam with them may be smaller than when cut-off is effected by the ordinary slide-valve and link-motion; they are, however, the cause of more loss than gain, and are now never used.

Hence, when only common valves and gear are to be used, the area for opening to steam when at its greatest should be such that the mean velocity of flow does not exceed 10,000 feet per minute. In actual practice the amount of opening is often much less than that given by the above rules, but it always results in loss of pressure in the cylinder throughout, and excessive "wire-drawing" previous to cut-off, and, therefore, should not be less but greater.

Exhaust Passages and Pipes.—The area of section of exhaust passages should be such that the mean velocity of steam does not exceed 6,000 feet per minute, and if the distance from the cylinder to the condenser is comparatively great, a much larger area is advisable. There should not be a greater difference than 1½ lbs. between the pressure in the cylinder and that in the condenser when exhausting, even with a high vacuum.

TABLE XXXI.

Piston Speed. Feet per Minute.	Diameter Main Steam ÷ D.	Diameter * Exhaust ÷ D.	Area Main Steam ÷ A.	Opening to Steam ÷ A.	Area * Exhaust ÷ A.
200	0·158	0·182	0·025	0·020	0·0333
250	0·177	0·204	0·0313	0·025	0·0417
300	0·194	0·223	0·0375	0·030	0·0500
350	0·209	0·241	0·0437	0·035	0·0583
400	0·224	0·258	0·0500	0·040	0·0667
450	0·237	0·274	0·0562	0·045	0·0750
500	0·250	0·288	0·0625	0·050	0·0833
550	0·262	0·302	0·0687	0·055	0·0917
600	0·274	0·316	0·0750	0·060	0·1000
650	0·285	0·329	0·0812	0·065	0·1083
700	0·296	0·341	0·0875	0·070	0·1167
750	0·306	0·353	0·0937	0·075	0·1250
800	0·316	0·365	0·1000	0·080	0·1333
850	0·326	0·376	0·1062	0·085	0·1417
900	0·335	0·387	0·1125	0·090	0·1500
950	0·344	0·397	0·1187	0·095	0·1583
1000	0·353	0·400	0·1250	0·100	0·1667

The exhaust passages from the high-pressure cylinder of a compound engine to the next cylinder should be such that the flow of steam does not exceed 4,500 feet per minute, in order that the difference between the pressure

* N.E. Coast Inst. E. and S. recommend for exhaust from H.P. cylinder 3,600 feet, from M.P. cylinder 3,900 feet, and from L.P. cylinder 4,500 feet, and difference in pressure 1¼ lbs. at working sea speeds.

during admission in the medium-pressure cylinder and exhaust in the high-pressure cylinder may not be excessive, and also from the medium-pressure to the low-pressure cylinder the nominal rate of exhaust should not exceed 4,500 feet. The pressure in the receivers is not sensibly constant, as it is in the condenser, being subject to sudden fluctuation when the high-pressure valve opens to exhaust and the medium-pressure valve opens to *lead*, and so on.

Table xxxi. gives the relation between the various passages, etc., and the piston in accordance with the foregoing rules, and is based on the assumption of a mean velocity of flow of 8,000 feet per minute for the main steam pipe, 10,000 for opening to steam, and 6,000 for exhaust; A is the area of piston, D its diameter, and ratio of D to length not more than 60.

In naval and other ships, the engines of which are only occasionally, and for short periods, run at full speed, so that high efficiency at those times is not of first consideration, the exhaust-pipe to the condenser may be of such a size when short that the flow is 7,300 feet in large engines and 6,500 feet in smaller ones; the exhaust-pipes from cylinder to cylinder may be correspondingly reduced, as may also the ports, etc.

TABLE XXXII.—WEIGHT OF STEAM IN LBS. AT 100 LBS. PRESSURE DELIVERED PER MINUTE THROUGH PIPES WITH A DROP OF 1 LB. ONLY.

By E. C. Sickles.

Diameter of Bore.	Length of Pipe.						
	25 Ft.	50 Ft.	75 Ft.	100 Ft.	125 Ft.	150 Ft.	175 Ft.
1 inch,	4·40	3·10	2·54	2·20	1·96	1·79	1·66
1¼ ,,	9·76	6·90	5·63	4·88	4·36	3·93	3·69
1½ ,,	13·04	9·29	7·69	6·52	5·83	5·33	4·95
2 ,,	30·54	21·60	17·60	15·20	13·60	12·50	11·58
2½ ,,	50·80	35·90	29·30	25·40	22·70	20·80	19·20
3 ,,	...	65·17	...	46·00	...	...	35·00
3½ ,,	...	98·30	...	69·50	...	...	52·70
4 ,,	...	138·1	...	97·60	...	...	73·80
4½ ,,	...	187·9	...	132·9	...	...	102·0
5 ,,	...	255·6	...	180·7	...	...	136·8
6 ,,	...	419·4	...	296·5	...	...	229·1
7 ,,	...	618·8	...	437·0	...	...	330·0
8 ,,	...	890·0	...	629·0	...	...	472·0
9 ,,	...	1206	...	553·0	...	...	644·0
10 ,,	...	1592	...	1126	...	...	851·0
11 ,,	...	2046	...	1447	...	...	1093
12 ,,	...	2575	...	1887	...	...	1428

Cylinder Liner.—In order that a suitable material may be supplied to resist the rubbing action of the piston without wearing away, and one that shall be capable of taking and retaining a polished surface, so as to minimise the friction of the piston, and which, when worn, may be easily and cheaply renewed, an inner bush or false barrel is fitted, usually called the *cylinder* *liner.* This liner should be made of a hard, close-grained metal having considerable strength, but not so hard as to resist the action of a cutting tool or file, and capable of taking and keeping a polish when rubbed by the piston-rings lubricated with soft water; it should also be such that the expansion caused by heat is practically the same as the cast iron of which the cylinder itself is made. It is usual to make these liners of cast iron, strengthened, closed, and hardened by mixing with it certain kinds of pig iron, or by the addition of a small quantity of steel (*vide* Chap. xxx.). The Admiralty prefer the liners to be made of steel, hammered or rolled to a proper size for machining; and some engineers use cast steel. Although the steel gives good results, it can be equalled by the specially-made cast iron, so far as good wearing is concerned, but, of course, steel exceeds cast iron in tensile strength; this latter quality was necessary to a higher degree for the horizontal engine than for the vertical engine; the Admiralty were justified in going to the expense of the steel, however, as it enable them to fit much lighter liners than would be admissible if made of cast iron. In the merchant service, with the vertical engine the cast-iron liner does exceedingly well, and it is not likely to be superseded by steel, even if this material can be manufactured much cheaper than at present. Liners are usually made with an inside flange at the bottom end (fig. 78), which fits into a recess in the cylinder end, and is secured there by *screw-bolts.* The upper end is turned for a few inches, so as to fit tightly into the cylinder shell at that part. The joint at the cylinder bottom is made with red-lead paint, while leakage between the liner and the cylinder shell is prevented at the other end by stuffing a few rounds of asbestos rope or Tuck's packing into a recess formed for that purpose, and preventing it from coming out by securing a flat wrought-iron ring *to the liner* so as to cover the packing. Sometimes in lieu of a stuffing-box, the outer edge of the liner and the edge of the turned part of the cylinder shell are champhered so as to form a groove; into this groove a turn of Tuck's packing or asbestos rope is pressed with a ring as before. Some engineers, preferring to rely on metallic contact, turn a slight recess instead of champhering the edge of the liner, and *caulk into it* a ring of soft copper. The Admiralty method of making a steam-tight joint at the outer end of the liner is by means of a flat copper ring covering the joint and secured to the liner, as also to the cylinder by means of iron rings and screw bolts. The copper ring is deeply grooved between these iron rings so as to permit of a slight movement of the liner endways with respect to the cylinder (*vide* fig. 78*a*). The liners are sometimes secured without a flange at the bottom, by screwing studs through the cylinder shell and liner, and making the ends steam-tight as before.

The space between the liner and shell should not be less than 1 inch, and may be filled with steam so as to prevent condensation in the cylinder, and heat the steam during expansion. If the cylinder has to be jacketed, this is really a better plan of doing it than by casting the cylinder and inner cylinder together, as was very generally done formerly. Independently of the advantage derived from the harder metal of which the liner may be made, compared with that which is suitable for so intricate a casting as a cylinder, there is another very great advantage to the manufacturer. Since it is a necessity that the inside walls of the cylinders be sound and free from sponginess, as well as blowholes, a casting may be condemned for a defect which in no way detracts from its strength or usefulness, excepting that it does not admit of the piston working on it steam-tight. If a liner is to be fitted, a little sponginess, or even a blowhole in the cylinder casting is of no consequence, and therefore the extra cost of fitting a liner does not, as a rule, exceed the reasonable premium which would be allowed for assuring good and sound castings; and this is especially so in the case of large cylinders.

False Faces.—For the same reason that liners are fitted to the cylinders, the cylinder face, on which the valve slides, needs a false face. This is usually made of hard, close-grained cast iron, of the same quality as the liner, and secured to the cylinder (fig. 78) by brass screws having *cheese* heads sunk

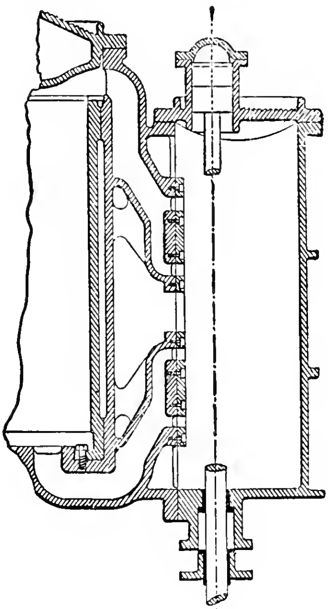


Fig. 78.—Section through Cylinder.

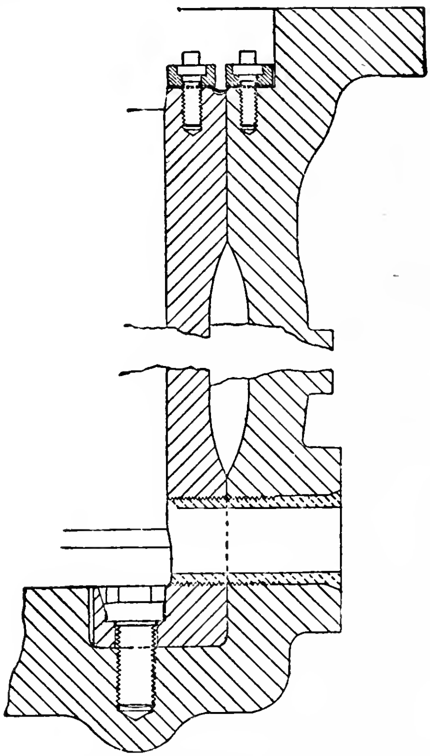


Fig. 78*a*.—Admiralty Method of Fitting Liners.

in a recess, so as to be considerably below the surface. Care should be taken to lock these screws, so that they cannot slack back; the simplest way of doing this is to cut a slight nick in the side of the recess, and caulk or drift the metal of the screwhead into it, after the screw is tightened in place.

False faces were sometimes made of hard gun-metal or phosphor-bronze in the engines of warships. The superior strength of these metals over cast iron admits of the face being much thinner, but besides being much more expensive, there is great risk of damage to the cylinder itself, owing to the greater expansion of the bronzes by heat: they never permit of such a good working surface as does cast iron ; and even if danger from this is slight, some difficulty has been experienced in keeping the joint between the two metals steam-tight.

By connecting the recesses for the screwheads with grooves cut in the face, the rubbing surfaces are well lubricated, and a considerable amount of relief given to the valve itself by the reduction of the effective pressure on it, caused by the steam flowing through these grooves, etc.

The corners of the ports, both in the false face and cylinder face, should be well rounded, as the casting is very apt to crack at them if they are sharp.

The Width of the Steam Ports, in the direction parallel to the cylinder bottom, is usually 0·6 to 0·8 of the diameter, but engines of longer stroke than usual require a larger proportion than this to obtain the necessary port area without having excessive length (measured in direction parallel to the axis). It is obvious that, at the cylinder bore, the width of port cannot exceed the diameter, and must really be somewhat less ; in actual practice it seldom exceeds 0·8 of the diameter ; but at the cylinder face it may, and sometimes does, exceed the diameter, the length being such that the area of section in the passage is uniform throughout.

Piston Valves.—If the very broad cylinder face is bent into the form of a cylinder, there will be the same area of orifice, while the space occupied in direction of the width of the port is less than one-third of that required for the flat face. The valve for such a face must be cylindrical, or composed of two circular discs or pistons having the same depth of edge as there would be of bearing surface at each end of the ordinary slide valve. Such a valve (fig. 138) is called a *piston valve*, and besides possessing the advantage of occupying little space, has the more valuable one of being free from lateral pressure, requiring no balancing or relief, and moving with the least resistance of any slide valve. For these reasons, the piston valve is an exceedingly good form when high pressures of steam are used and for very large engines. It is a very general thing for engineers to fit a piston valve to the high-pressure cylinder of compound engines of all sizes and types ; many makers also fit the medium-pressure cylinder with piston valves, and a few fit them to all three cylinders, especially of large engines, and of engines running at high revolutions, as in torpedo boat destroyers. The earliest compound engines of Woolf, made in the early years of the nineteenth century, had piston valves.

"**Drop**" **Valves,** with some form of Corliss valve gear, may take the place of piston valves in larger engines, at least if superheated steam comes into more general use. They are extensively used on the large slow-speed marine type of engine on shore, and in the slow revolution paddle steamer in America (*v.* fig. 30, p. 73).

Double-Ported Valves.—Although there is of necessity only one opening of the steam passage into the cylinder, there may be two or more openings *through the cylinder face* into the steam passage. The combined area of these openings need not materially exceed that of the section of passage, but it is better to make it so ; as each will be open to steam by the same amount as the single port, if the valve has the same travel, lap, etc., the total opening is in this case double that of the single port for a double-ported face, and treble for a treble-ported face. When the face is treble-ported, the valve is generally arranged so as to admit steam through all three ports, but to exhaust through two only, as there is seldom any difficulty in getting full opening to exhaust.

Steam Jackets.—It is not necessary here to enter into the question of the economy of steam-jacketing. If the economy is doubted, it is, at least, certain that it admits of the cylinders being gently warmed before starting, without moving the working parts. It was customary at one time to form a jacket around the cylinder by *casting it* with two thicknesses of metal; but this was often inconvenient, and always risky to the moulder. Now, as a loose liner is fitted, the jacket is automatically provided. When the engines are of short stroke, the ends present almost as large a surface to the steam as do the walls, and should be steam-heated if the jacketing is to be effective. For strength of structure, too, all large cylinders should have hollow bottoms and covers, as stiffening them with ordinary webs is not always sufficient, and is in some cases of iron castings even a source of danger. If the pressure is on the same side as the webs, they are safe and do add to the strength of structure; if on the opposite side, then they are in tension, and their outer edge liable to extreme tension; so that if there be a nick or other such defect from which to start a crack, or if subject to a sudden application of the stress, the outer edge is apt to crack, which will develop and spread into the main body of metal, finally causing serious damage. This arises from the fact of cast iron possessing so low a power of resisting stress in tension, compared with its power against compression. Care should be taken to thoroughly drain the steam jackets, and to this end no webs should so be placed as to stop the flow of water to the drain-cocks. Steam, when supplied to the jackets of the low-pressure cylinder, should not much exceed the pressure that is in the receiver; for this purpose, a reducing valve is fitted between the boiler supply pipes and the jacket.

Boring Holes.—The diameter of the boring hole depends generally on the size of the boring bar employed, and should not be less than one-fifth the diameter of the cylinder. When there is a single piston-rod, the stuffing-box is formed in the cover of the boring hole. When there are two or more piston-rods, the boring hole should be large enough to admit a man; and, therefore, not less than 14 inches diameter, and, when possible, 16 inches diameter. The doors are sometimes made with the flange fitting into a recess *inside* the cylinder, so that the piston-rod, having a butt end, may be drawn from the cylinder with the piston; when this is required, the boring hole must be of sufficient diameter to admit of the piston-rod end drawing through it.

Auxiliary Valves.—To render engines handy, so that they may be started from any position of the cranks, it is necessary to arrange the gear so that the main valves may be operated by hand, or else to fit smaller valves, which may be readily worked by the engineer; these are called auxiliary valves, and should have a port area equal to 0·002 the area of the piston. It is usual to fit such valves to the low-pressure cylinder of compound engines. Pass cocks, by which steam can be admitted to the valve boxes of intermediate cylinders, should be fitted, and by their means the use of the L.P. auxiliary valve is often avoided.

These auxiliary valves are usually only flat plates, without even an exhaust cavity; but for very large engines they should be piston valves, or some form of balanced valve.

Escape or Relief Valves.—These are simply spring-loaded safety valves, to allow of the escape of water caused by priming or condensation when the piston drives it to one end of the cylinder. They are fitted to each end of the high-pressure cylinders of all fast-running marine engines. It is sufficient in vertical engines if there is only an escape valve at the bottom of the medium-pressure and low-pressure cylinders. The diameter of these valves should be one-fifteenth the diameter of the L.P. cylinder of a compound engine. In the Navy it is usual for all large cylinders to have a *pair* of escape valves at each end.

Drain-Cocks.—These should be placed wherever any water is likely to accumulate in the cylinder and casings, and their size 0·023 the diameter of cylinder $+ \frac{1}{4}$ inch. They should be connected to a pipe leading into the condenser bottom; for if led to the bilge the engine-room is filled with steam when open, and the receiver and low-pressure cylinder will seldom drain—in fact, during the greater part of the stroke, instead of letting water out, they let air into the low-pressure cylinder and spoil the vacuum. Care should be taken when the drain-pipe is led to the condenser that the water, etc., does not impinge on the tubes, or even on the condenser sides, so as to do serious damage, and a non-return valve fitted to prevent water getting back.

Receiver Space.—The space between the valve of the high-pressure cylinder and that of the medium-pressure cylinder, and that between the valves of the medium-pressure and the low-pressure cylinders, should be from 0·6 to 1·1 times the capacity of the exhausting cylinder, when the cranks are set at an angle of 120°. When the cranks are opposite or nearly so, this space may be very much reduced. The pressure in the medium-pressure receiver should never exceed 0·7 the boiler pressure, and is generally much lower than this. It is usual to fit a safety valve to the low-pressure receiver, loaded by weight or spring to a pressure of 20 to 30 lbs. per square inch; otherwise, owing to the large flat sides between the two cylinders, and in the valve-box when a flat valve is employed, great risk of explosion would be run. This safety valve is usually of the same size and design as the cylinder escape valves. The receivers of three-crank compound engines need not be so large as the above, as the cranks are usually at angles of 120°; in the case of triple-compound engines with the medium-pressure leading the high-pressure, a smaller receiver will do.

Column Facings and Feet.—It was very usual at one time to form merely facings for the jointing of the cylinder to the frames and columns; but as this necessitated the use of studs, or else driven bolts with the heads inside the cylinders, it is now abandoned, distinct projections or feet being cast to the cylinder bottom, having flanges corresponding to those on the columns or frames, so that they may be connected by driven bolts, which are always accessible. The only objections to this method are, that it is more expensive to mould, and a certain amount of risk is run of getting the casting sound and strong where the feet meet the main casting. The former should be disregarded in considering so important a part as the cylinder, and the latter is always avoided by a good moulder.

Great care should be exercised in designing these column feet, for through them the load due to the steam on the cover (which is greater than that on the piston) is transmitted; and as the load is a recurrent one and always applied suddenly, very ample section of metal should be provided to sustain it. The area of section through these feet should be such that the stress does not exceed 600 lbs. per square inch—that is, the area in square inches is not less than that given by dividing the *maximum* load on the cover in pounds by 600. The webs from the flanges of the feet should be well spread over the cylinder bottom and towards the sides, so as to distribute the strain.

Holding-down Bolts.—The bolts connecting the cylinder to the columns or frames should be such that the stress on them does not exceed 4,000 lbs. per square inch, taking the section at the bottom of the thread, and when there is a large number of comparatively small size it should not exceed 3,000 lbs. per square inch (*v.* Chap. xxviii.).

Horizontal Cylinders.—In addition to the facings or feet for connecting to frames, additional feet were necessary for the cylinders of horizontal engines to rest on and be secured to the engine bed. These feet, too, had webs so arranged as to distribute the strain caused by the reaction from the weight of the cylinder, pistons, etc. The front part of the cylinder was rigidly bolted down, while the back end, especially of long cylinders, was *held down* only, and free to move horizontally when expanded by the heat. But since cast iron will expand only one-tenth of an inch in 8 feet, by an increase of 180° Fahr. of temperature, there was seldom need to make any special provision, beyond boring the holes for the back bolts rather larger, or making them slightly oval in the cylinder feet.

Oscillating Cylinders.—The chief peculiarity of these cylinders is the method of supporting by trunnions, which also serve as steam and exhaust-pipes (*vide* fig. 27). Half the load on the piston is taken on each trunnion; and since they are of such ample diameter, it is sufficient to assume that the metal is subject only to shearing stresses, and therefore the area of section should be such that the stress does not exceed 950 lbs. per square inch. The diameter of the trunnions is governed by the size of the exhaust-pipe, since the steam must exhaust through one of them, and it is usual and convenient to make them both of the same size. In the case of a compound oscillating engine, the trunnions of both cylinders should be of the same size, which will depend on the size of exhaust of the low-pressure cylinder. The trunnions of the high-pressure cylinder, being so much larger than is necessary to accommodate the steam-pipe, allow of a space between the outer or working part and the inner part or stuffing-box, which, if left open to the air, is well ventilated, and so reduces the heat on the bearing due to steam of high temperature.

The *length* of the trunnion journal or bearing should be such that the pressure per square inch on the area, made by the multiple of its diameter and length, does not exceed 350 lbs.; generally it is from one-third to one-half of the diameter.

The trunnions have interposed between them and the cylinder body a belt, which conveys the steam to and from the valve-boxes. This belt should be very strong and well ribbed to the body of the cylinder, immediately above and below the trunnions, and when the cylinder is fitted with a liner, it is better to form the outer shell in the shape of a beer barrel, so that the belt projects inside and not outside, as it would be were there no liner; the strain from the trunnions is then at once taken by the cylinder sides without the intervention of webs and ribs.

The cylinder valve faces of oscillating engines should be set so that the edge next the steam entrance should be the nearest point to the cylinder—that is, the plane of the cylinder face touches a cylinder whose axis coincides with the axis of the cylinder-bore at this edge. When this is so, the lead of the steamway into the valve-box is short and easy, and the opening into the exhaust-belt on the side opposite is large, without causing the valve-spindle centre to be unnecessarily far out from the cylinder.

Cylinder Covers.—Like the cylinder end or bottom, the cover has to be strong enough to take the full steam pressure, but as a rule it has no load to distribute to any other part. The same remarks as to webs, etc., equally apply to the covers, and all above 24 inches diameter for high-pressure cylinders, and 40 inches diameter for low-pressure cylinders, should be made hollow with two thicknesses of metal. Those of vertical engines are better made in that way for all sizes, inasmuch as it is necessary to fill in the spaces between the webs, when they are so made, to prevent the lodgment of water, etc., and it is usual to add a false cover, either polished or cast with a pattern to give a good appearance. This can always be accomplished by casting the covers hollow. The cylinder covers of naval engines and for large express steamers are made of cast steel, and formed with ribs having bull-nosed flanges to strengthen them. In small naval engines, such as those of *destroyers*, the covers, especially of the medium-pressure and low-pressure cylinders, are often made of manganese or other strong bronze, which has an elastic limit higher than that of ordinary steel, and permits of them being cast much thinner and lighter. Covers, when of steel or bronze, are sometimes corrugated as well as coned to get the necessary stiffness without the complications of ribs. The depth of the cylinder cover at the middle should be about one-quarter of the diameter of the piston for pressures of 80 lbs. and upwards; that of the low-pressure cylinder cover of a compound engine should be at least 0·15 its diameter. Since, however, the size of the piston-rod is the best measure of the pressure on the cover, it is better so to design the cover that its depth at the middle is not less than 1·45 times the diameter of the piston-rod. The depth of the cover at the edge depends on the steam port; a recess being formed for the steam-way, and the inside of the cover otherwise being parallel to the piston.

It is the custom with some engineers to end the cylinder a little beyond the extreme travel of the piston, the steam port-opening being then in the same plane with the cylinder flange; the cover has a large recess in it, and its flange so extended as to enclose the port-opening (fig. 79). The advantage of this method is the decreased length and weight of cylinder, and being able to secure the cover in way of the steam port direct to the main casting, instead of to the comparatively weak bridge of metal across the port. On the other hand, however, the cover occupies considerably more room, and not being of circular form at the flange, cannot be turned so easily. For larger engines this plan is a very good one, and may be adopted with advantage, but for small ones and those of moderate size it is not always so convenient as the older one.

Cylinder Cover Studs and Bolts should be made of the best steel, and of such a size that the stress on them does not exceed 5,000 lbs. per square inch of section at the bottom of the thread, as they are subject to sudden and intermittent loads and severe and sudden shocks when *priming* occurs, also to considerable wear and tear from the frequent removals of the covers for examination of the pistons. In large engines it is usual to fit the cylinder covers with manholes for purposes of examination; as for such engines larger studs may be fitted, a higher stress is permissible if desired, so that the section may be such that with the maximum pressure there is a stress of 5,500 lbs. per square inch. From one or two causes the resistance to pressure on the cover may not be evenly distributed over the whole of the studs, so that a good nominal margin of safety should be allowed in such a very important part; this is especially so when a large number of small studs are fitted; in this case, when of less diameter than $1\frac{1}{8}$ inch, the allowance should not exceed 4,500 lbs. per square inch; under $\frac{7}{8}$ inch no more than 3,500 lbs.

Cylinder Flanges.—The width of the cylinder flange need not exceed three times the diameter of the bolts or studs, but if the former are fitted this allowance is not sufficient to give space for the heads. Studs are now nearly always fitted to marine cylinders in great measure for this reason.

Clearance of Piston.—If both cylinder end, piston, and cover were accurately turned, as is very desirable and required by the Admiralty, and the brasses did not wear, a very small amount of space would suffice for clearance between the piston and cylinder end; but as it is usual to leave these parts as they come from the foundry, and the bearings, however well made, may wear in course of time, it is necessary to make due allowance for this. Small engines up to 50 N.H.P. require an allowance of $\frac{1}{8}$ inch at each end for roughness of castings, and $\frac{1}{16}$ inch for each working joint—that is, for any part between the piston and the shaft journals where wear can take place; engines from 50 N.H.P. to 100 N.H.P., $\frac{3}{16}$ inch and $\frac{3}{32}$ inch for each working part; engines from 100 N.H.P. to 200 N.H.P., $\frac{1}{4}$ inch and $\frac{3}{32}$ inch for each working part; engines over 200 N.H.P., $\frac{3}{8}$ inch and $\frac{1}{8}$ inch for each working part. Fast running engines should have a larger allowance, or be turned as above recommended. This allowance is usually called the ***stroke clearance.***

For example, take the case of a vertical direct-acting engine of 120 N.H.P.; the parts which wear so as to bring the piston nearer to the bottom are three—viz., the shaft journals, crank-pin brasses, and piston-rod gudgeon brasses, so that the clearance at *top* will be $\frac{3}{8}$ inch, and the clearance at *bottom* $\frac{3}{8}$ inch $+ 3 \times \frac{1}{8}$, or $\frac{3}{4}$ inch in all. Here the total clearance in the cylinder is $1\frac{1}{8}$ inch.

Valve-Box Covers.—The covers are usually formed of a flat plate stiffened by ribs or webs. So long as the stiffening webs are on the same side as the pressure this form is satisfactory, especially when the covers are not very large; but when they are large and it is inconvenient to web them on that side they should be made hollow, with the back rounded like a hog-back girder. These covers in Naval ships and express steamers are now very generally made of cast steel.

The studs and bolts with which these doors are secured should be arranged in accordance with the rules laid down for those of cylinder covers.

Small Doors and Covers.—As it is essential to examine from time to time the internal working parts, and as this examination is more for the sake of seeing that everything is in good order, rather than in the expectation of having to execute repairs, and generally there is little time at the disposal of the engineer for these purposes, small doors should be fitted to the large heavy doors, secured with only a few studs, so as to be quickly taken off and refitted. These may for this reason, with advantage, be of cast steel or of steel plate pressed to shape. There should be, when possible, a doorway in the bottom and cover, and to the valve casing of the low-pressure cylinder, large enough to admit a man. To the valve-boxes of all cylinders there should be peep-holes, through which to ascertain the leads and cut-off of the valves, and to press the flat valves to the cylinder faces should they have become blown off.

Lagging and Clothing of Cylinders.—All hot surfaces, from which loss of heat may arise by radiation, should be covered with a non-conducting substance. Felt was formerly employed and well suited for this purpose, being enclosed in polished teak or mahogany lagging, secured with brass bands wherever in view in the engine-room, and by pine lagging or canvas when not in view. Sheet-steel is, however, now used as being more enduring than wood, and when carefully fitted and highly "barfed" looks as well as polished wood, and it lasts much longer.

Cement of kinds, silicate cotton, and asbestos are specified by engineers for the cylinder covering, but silicate is very objectionable, on the ground that the dust coming from it through the lagging, owing to the vibration, etc., is

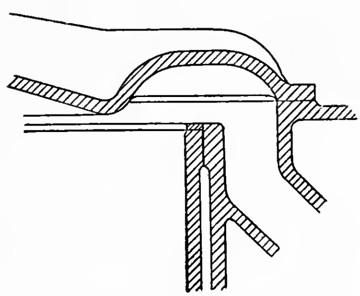


Fig. 79.—Shortened Cylinder with Port in Cover.

very apt to get into the bearings and guides, and cause serious trouble. Asbestos fibre and magnesia may, however, be used with advantage, especially on the high-pressure cylinders; a cement made of this fibre, etc., has been found very successful; a cement, consisting largely of carbonate of magnesia, has also proved an excellent covering for these and other hot surfaces. It is very necessary to cover up *all* hot surfaces when using steam of such high temperatures as are now common, and considerable gain is obtained by putting muffles on the cylinder covers and all metallic-exposed surfaces. The value of the different non-conducting materials are given in Chap. xxvi.

The following rules are for the scantlings of the cylinder and its connections:—

D is the diameter of the cylinder in inches.
p the load on the safety-valves in lbs. per square inch.
p_1 the absolute pressure of steam at H.P. valve-box.
f a constant multiplier = thickness of barrel + ·25 inch.

Thickness of metal of cylinder barrel or liner, not to be less than $p \times D \div 3000$ when of cast iron.*

$$\text{Thickness of cylinder barrel} = \frac{D}{6000}(p + 50) + 0{\cdot}2.$$

$$\text{,, liner} = 0{\cdot}8 \times f.$$

For purposes of calculation p may be taken as follows:—

H.P. cylinder	p = boiler pressure, or $p_1 - 15$.		
M.P. ,,	triple p	= 0·6 ×	boiler pressure
M.P. ,,	quadruple p	= 0·7 ×	,,
$M.P._2$,,	,, p	= 0·45 ×	,,
L.P. ,,	,, p	= 0·22 ×	,,
L.P. ,,	triple p	= 0·25 ×	,,

Thickness of liner when of steel	= 0·65 × f.
,, metal of steam ports	= 0·6 × f.
,, ,, valve-box sides	= 0·65 × f.
,, ,, ,, covers	= 0·7 × f.
,, ,, cylinder bottom	= 1·1 × f, if single thickness.
,, ,, ,, ,,	= 0·65 × f, if double ,,
,, ,, ,, covers	= 1·0 × f, if single ,,
,, ,, ,, ,,	= 0·6 × f, if double ,,
,, cylinder flange	= 1·4 × f.
,, ,, cover flange	= 1·3 × f.
,, ,, valve-box flange	= 1·0 × f.
,, ,, door flange	= 0·9 × f.
,, ,, face over ports	= 1·2 × f.
,, ,, ,, ,,	= 1·0 × f, when there is a false face.
,, ,, false face	= 0·8 × f, when cast iron.

For torpedo boats and gunboats, destroyers, and other such ships where extreme lightness is a necessity, and full power is only developed at intervals and for a short time, the scantling may be reduced by 25 per cent.

Pitch of Studs or bolts in cylinder-cover or valve-box door in inches should not exceed $\sqrt{\frac{t \times 130}{p}}$, t being the thickness of the cover or door flange in sixteenths of an inch, p, the pressure per square inch in pounds on it.

Flat Surfaces.—All flat surfaces of cast iron should be stiffened by webs, or stays of some form, whose pitch should not exceed $\sqrt{\frac{t^2 \times 50}{p}}$. These webs should be of the same thickness as the flat surface, and their depth at least 2·5 times the thickness.

The L.P. Cylinder Body or Barrel should be stiffened by external flanges or webs, at about 12 times the thickness of metal apart; these webs should be 1·5 × f thick, and stand at least 0·75 × f beyond the surface of the cylinder. Some engineers, however, prefer to do without these stiffening webs, and make the cylinder somewhat thicker instead. The low-pressure cylinder, however, differs from the others in size, and in being exposed to external pressure in excess of the internal; therefore it should have webs, especially when of large diameter.

* When made of exceedingly good material, at least twice melted, the thickness may be 0·8 of that given by the above rules.

Stuffing-Boxes and Glands.—For obvious reasons, it is useless giving any definite rules for the sizes of these, as they will differ from different circumstances, and many of the parts do not vary when others are varied.

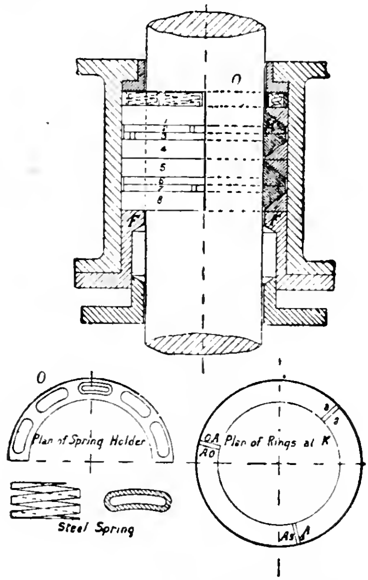


Fig. 80.—Combination Metallic Packing.

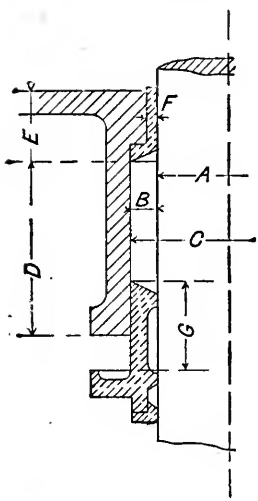


Fig. 80*a*.

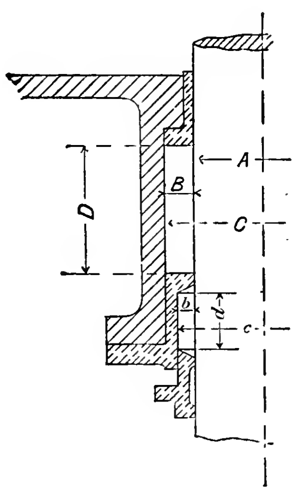


Fig. 80*b*.

The following Table of sizes gives such as are found in good practice, and will be of more use than any abstract rules :—

STUFFING-BOXES, etc.

TABLE XXXIII.—STUFFING-BOXES FOR ELASTIC PACKING (Fig. 80a).

A	B	C	D_1 50 lbs.	D_2 100 lbs.	D_3 150 lbs.	E	F	
Diameter of Rod.	Width of Packing Space.	Diameter of Box. 1·25 × A + ·55.	Depth of Packing. ·9 × A + ·75.	Depth of Packing. 1·15 × A + ·75.	Depth of Packing. 1·4 × A + ·75.	Overall Depth of Neck-Ring. ·45 A + ·35.	Thickness of Bushes, Gland, and Neck. ·045 × A + ·18.	Number and Diameter of Studs.
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
3/4	3/8	1 1/2	1 3/8	1 5/8	1 3/4	3/4	3/16	2 of 3/8
7/8	3/8	1 5/8	1 1/2	1 3/4	2	3/4	3/16	2 ,, 3/8
1	7/16	1 7/8	1 5/8	2	2 1/4	7/8	1/4	2 ,, 1/2
1 1/8	7/16	2	1 3/4	2 1/8	2 3/8	7/8	1/4	2 ,, 1/2
1 1/4	7/16	2 1/8	1 7/8	2 1/4	2 1/2	7/8	1/4	2 ,, 1/2
1 3/8	7/16	2 1/4	2	2 3/8	2 5/8	1	1/4	2 ,, 5/8
1 1/2	1/2	2 1/2	2 1/8	2 1/2	2 7/8	1	1/4	2 ,, 5/8
1 3/4	1/2	2 3/4	2 1/4	2 3/4	3 1/4	1 1/8	1/4	2 ,, 3/4
2	9/16	3 1/8	2 1/2	3 1/8	3 1/2	1 1/4	1/4	2 ,, 3/4
2 1/4	9/16	3 3/8	2 3/4	3 1/2	3 7/8	1 3/8	5/16	2 ,, 7/8
2 1/2	5/8	3 3/4	3	3 3/4	4 1/4	1 1/2	5/16	2 ,, 7/8 3 ,, 3/4
2 3/4	5/8	4	3 1/4	4	4 5/8	1 5/8	5/16	2 ,, 1 3 ,, 3/4
3	11/16	4 3/8	3 1/2	4 1/4	5	1 3/4	5/16	2 ,, 1 3 ,, 7/8
3 1/4	11/16	4 5/8	3 3/4	4 3/4	5 3/8	1 7/8	3/8	2 ,, 1 1/8 3 ,, 7/8
3 1/2	3/4	5	4	5	5 5/8	2	3/8	2 ,, 1 1/8 3 ,, 1
3 3/4	3/4	5 1/4	4 1/4	5 1/4	6	2 1/8	3/8	2 ,, 1 1/4 3 ,, 1
4	13/16	5 5/8	4 1/2	5 1/2	6 3/8	2 1/8	3/8	2 ,, 1 1/4 3 ,, 1
4 1/2	7/8	6 1/4	4 3/4	6	7	2 3/8	7/16	2 ,, 1 3/8 3 ,, 1 1/8
5	15/16	6 7/8	5	6 1/2	7 3/4	2 5/8	7/16	2 ,, 1 3/8 3 ,, 1 1/8
5 1/2	1	7 1/2	5 5/8	7	8 1/2	2 7/8	7/16	4 ,, 1 1/8
6	1 1/8	8 1/4	6 1/4	7 5/8	9 1/4	3	7/16	4 ,, 1 1/8
6 1/2	1 1/8	8 3/4	6 5/8	8 1/8	9 7/8	3 1/4	1/2	4 ,, 1 1/8
7	1 1/4	9 1/2	7	8 3/4	10 1/2	3 1/2	1/2	4 ,, 1 1/4
7 1/2	1 1/4	10	7 1/2	9 3/8	11 1/4	3 3/4	1/2	4 ,, 1 1/4
8	1 3/8	10 3/4	8	10	12	4	9/16	4 ,, 1 1/4
8 1/2	1 3/8	11 1/4	8 1/2	10 1/2	12 3/4	4 1/4	9/16	4 ,, 1 3/8
9	1 1/2	12	8 7/8	11	13 3/8	4 3/8	9/16	4 ,, 1 3/8
9 1/2	1 1/2	12 1/2	9 3/8	11 5/8	14 1/8	4 5/8	5/8	4 ,, 1 3/8
10	1 1/2	13	9 3/4	12 1/4	14 3/4	4 7/8	5/8	4 ,, 1 3/8

In the case of the cylinder, it is usual to make the stuffing-boxes, for uniformity's sake, for the low-pressure and medium-pressure piston-rods of the same depth as that of the high-pressure rod. The stuffing-boxes of the valve-spindles, too, are usually exceptionally deep, on account of their liability to leak, and the trouble of packing them. The packing of the stuffing-boxes when in the cylinder covers of vertical engines is very liable to give trouble with steam of high pressure from the want of moisture; the lubricant affects only the top layers of packing, and keeps them soft, while the bottom ones get hard and charred. Metallic packings (v. fig. 80) are the best for use with steam of high-pressure, and although some patent vegetable packings work very well, these latter are gradually being superseded by the former. The metallic packings are generally arranged in a series of hoops of triangular section, the pressure on the rod being caused either by a second set of hoops outside the first, causing a wedging action on the gland being pressed home, or else by an arrangement of springs or spring clips. The newer and better forms are now giving great satisfaction, and have taken the place of vegetable and asbestos in the high-pressure cylinder, and the medium-pressure cylinder also. Many engineers fit metallic packing in the low-pressure cylinder glands, others strongly object to do so, much preferring good vegetable packing on account of the excessive moisture in this cylinder.

Metallic packing is, however, that now always used in marine engines of 50 N.H.P. and upwards for piston-rods, and generally for all rods exposed to steam above 3 inches diameter.

CHAPTER XI.

THE PISTON—PISTON-ROD—CONNECTING ROD.

The Piston is essentially only a disc, strong and stiff enough structurally to withstand the pressure of the steam on it, and fitting steam-tight in the cylinder. The piston in this simple form is seen in the Richard's Indicator, and is often so fitted to small engines.

In the early days of steam-engine construction, when there existed no machine capable of truly boring out a cylinder, the bore was not perfectly cylindrical, nor the sides very smooth, and, consequently, unless some form of elastic packing was interposed between the piston and the cylinder sides, it could not work steam-tight. It was customary to form the piston with a recess on the rim, into which rope or *junk* was coiled, just as it was usual to do with all pumps. This packing could not be examined or renewed without drawing the piston from the cylinder, a tedious operation at all times; to remedy this the recess was made without a flange at the top or side of the piston next the cylinder cover, but a false flange or loose ring was bolted to the piston so as to retain the junk packing in place, and admit of its being removed or added to without removing the piston from the cylinder. This ring was called the "junk-ring," and retains that name, although junk is no longer used to pack pistons. After a few weeks' work the cylinder was rubbed smooth and fairly true, when the piston would work steam-tight with very little friction, and with moist steam of low-pressure and temperature the packing lasted a considerable time.

A solid piston—that is, one without packing—is really the best for good working, so long as it remains steam-tight; but as there is always some slight amount of wear, especially when the cylinder is fresh from the boring mill, and leakage past the piston is most serious, more particularly when the engine is standing still, it is necessary to have some means of adjustment, whereby the piston is maintained a steam-tight fit in the cylinder.

In lieu of the vegetable packing, which is not admissible with steam of high pressure, engineers now fit metallic rings, called "packing-rings," in various forms, which are pressed outwards against the side of the cylinder by their own elasticity or by springs. These rings are maintained in position steam-tight by the junk-ring as of old.

When these metallic rings are once in place so as to fit closely to the cylinder sides, there is no need of further lateral pressure until by wear the piston becomes slack, and steam permitted to pass it. However, nearly all existing pistons are automatic in this respect, and the consequence is that the packing-rings press so tightly on the cylinder sides, that the loss by friction seriously impairs the efficiency of the engine; and it is only when the ring or cylinder is considerably rubbed away, that the piston works with ease. From these causes many really good pistons have been condemned after *having been made* to cause serious damage.

Perhaps the first remove from the primitive piston is to be found in the form usually fitted in locomotives, and generally known as *Ramsbottom's*.

Ramsbottom's Rings (fig. 81).—The late Mr. Ramsbottom, of the L. & N.W.R. Co., was the first to pack pistons by one or more narrow metal rings. turned somewhat larger in external diameter than that of the cylinder bore. and which, after being cut across so as to be capable of being compressed to suit the bore of the cylinder, are fitted into recesses turned in the piston edge. The rings fit easily into these recesses, and as they are so placed that no two of the joints are in a line, the piston is practically steam-tight, and works very well in locomotives and other quick-working engines of small size; but for large engines, and engines undergoing the same vicissitudes as those on shipboard, there is an objection to this form of piston. It will be seen that the rings cannot be examined or removed without drawing the piston, and that there is no means of preventing steam from passing where the spring is cut across, besides which the rubbing surface is very small, and the spring is always exerting its maximum effort. The first of these objections is overcome (fig. 81*a*) by fitting a junk-ring, having cast with it a spigot or ring, which goes down into the recess around the piston for the packing-ring, and made steam-tight; into grooves turned in the outer surface of this spigot the Ramsbottom rings are fitted.

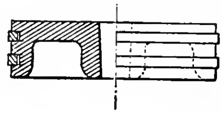
For small engines these rings are made of steel; for such engines as may be standing unused for many days, engineers prefer to fit hard bronze rings. For larger engines, where the section may be three-quarters of an inch square and upwards, the rings are better of tough and hard cast iron, which works very well indeed.

Common Piston-Rings (figs. 82, 85*a*, 86, and 87) consist only of a single hoop made of very tough, close-grained, cast iron, made on the same principle as the Ramsbottom rings, but fitted steam-tight between the piston flange and the junk-ring, and free to move laterally. This packing ring is usually turned to a diameter about 1 per cent. in excess of that of the cylinder, and either cut across diagonally, or formed so that one end has a tongue fitting into a recess in the other (fig. 87), a brass cover-piece being fitted behind the gap, so as to prevent steam leaking into the space behind the rings. The ring is then fitted to the piston flange steam-tight by scraping both surfaces; the ring is raised by interposing very thin pieces of paper between it and the flange, and the junk-ring is then fitted steam-tight to the piston and packing-ring by scraping, etc. Some makers of pistons profess to turn the piston and rings so accurately as to require no scraping, and with the perfection of the modern lathe and the high-class tool steel now used this is possible with care; other engineers prefer to grind the rings tight after coming from the lathe. In whatever way the object is attained is of small moment compared with the necessity of having the ring *perfectly* steam-tight between the flange and junk-ring.

Piston Springs.—When the piston is of comparatively small diameter, the elasticity of the packing-ring itself is sufficient to keep it steam-tight against the cylinder sides for a very considerable time after it is fitted; and even larger rings may be made of sufficient strength to do this, but they would then be open to the same objection as raised against the Ramsbottom rings.

The old method of pressing the ring out by means of dished springs or coach-springs, as shown in fig. 85*a*, is now seldom used in new engines; the objections to it are the uneven and unknown pressure exerted, and the reaction of the piston itself, from the fact of the springs pressing on it. It was a very difficult thing to set every spring so that the pressure on the ring was uniform;

Fig. 81.



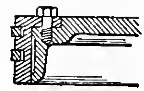
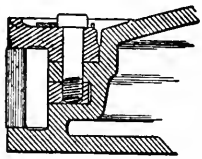


Fig. 81*a*.

Fig. 82.



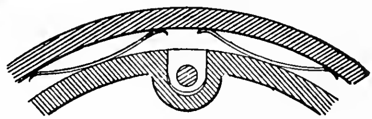
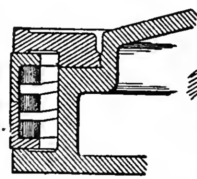


Fig. 85*a*.

Fig. 83.



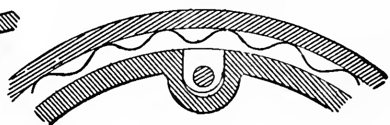


Fig. 86.

Fig. 84.

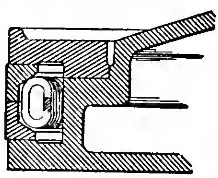
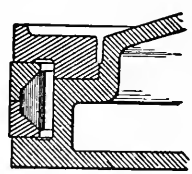


Fig. 85.



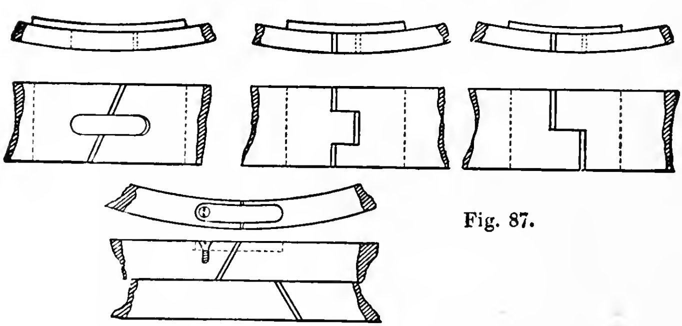


Fig. 87.

Figs. 81 to 87.—Various Forms of Piston Packings.

and the range of action of this form of spring is very limited, so that, although the ring might be very tight when first fitted, after a few days running it might be passing steam. The surface taking the pressure too was small, and the springs were apt to bed themselves into the ring, and in doing so wear through their curved ends. These defects were partially remedied by adding to each one or more subsidiary springs on the principle of coach-springs, but that only tended to aggravate the other evil spoken of—viz., the reaction of the piston itself.

When a piston is moving through its course, and guided therein by the rod at one end and the tail rod (or back guides in case of a horizontal engine) at the other, it should be quite free laterally from the packing-ring, which may follow its course freely. When the bore of the cylinder is quite true, and its axis coincides with the line of motion of the piston centre, it is of no consequence if the springs do bear on the piston; but if the cylinder wears somewhat out of truth in either direction, it is important that the spring-ring shall follow the sides of the cylinder freely; it cannot do this if the springs react from the piston body.

Cameron's Patent.—Fig. 86 shows a piston-ring pressed out with a corrugated ribbon of steel; the lateral pressure here is obtained by the resistance of the spring to being bent into a circle, and by the pressure exerted by the corrugations when the ends of the spring are pressed apart. This spring exerts an almost uniform lateral pressure on the packing-ring without touching the body of the piston, and by making the packing-ring comparatively thin, it will adapt itself to the shape of the cylinder when worn. The pressure on the ring can also be easily and nicely adjusted by packing pieces between the ends of the spring. One great advantage of this spring is that it can be fitted to any piston without condemning any of the parts beyond the springs.

Mather and Platt's Patent.—It was found that metallic packing-rings not only wore sideways, but also on the edges, so as to become slack between the flange and junk-ring; a very slight amount of play with heavy rings causes a very large degree of slackness from the continual concussion on change of motion at every stroke. To obviate this the ring was formed with inside flanges, as shown in fig. 83, and split into two, a spiral hoop, having three or four turns, being coiled inside the rings, whose action is to press the packing-rings outward against the cylinder sides, and up and down against the flange and junk-rings. This form has been generally very successful, and pistons so fitted have worked very well indeed; but there is the objection that no adjustment of the spring is possible, and it is always exerting its maximum effort. The chief part of the elasticity of the spring, however, is exerted in pressing the rings against the flange and junk-ring, and the friction so caused helps to prevent undue pressure on the cylinder side, so that in practice it is not found that there is excessive side pressure when first fitted, nor lack of it when the cylinder is worn. These springs are made of steel, or very strong cast iron, cut out of a ring of either metal. They are also sometimes cast to the form required.

Buckley's Patent consists of two rings, of section as shown in fig. 84; a spiral coil of steel wire is bent into a circle, and inserted between the two packing-rings. Pressure is exerted in the same way as in Cameron's spring, and tends to press the packing-rings both outwards and against the flange and junk-ring. This form of piston is still often used at the present time, and when properly adjusted works very well. The spring, however, is a very stiff one, and requires but little end displacement to exert a very considerable pressure on the sides of the cylinder.

Qualter and Hall's Patent.—This piston has two packing-rings of triangular section, with a third ring inside and between them, as shown in fig. 85, so that on this inner ring being pressed outwards, it exerts a wedge action on the two packing-rings so as to force them against the flange and junk-ring. Coach-springs are employed to press the ring outwards, which are each held in a brass frame having a tapered piece on the back, which fits into a recess in the piston and against a tapered cotter; this cotter can be pressed down by a set screw in the junk-ring, and any required pressure is imparted to the ring through the spring in this way. This piston has, therefore, the advantage of being capable of adjustment without removing the junk-ring, and the adjustment can be made to a nicety with very little trouble, and also, in case of the engines not being required for some weeks, the pressure on the springs may be relieved until required again.

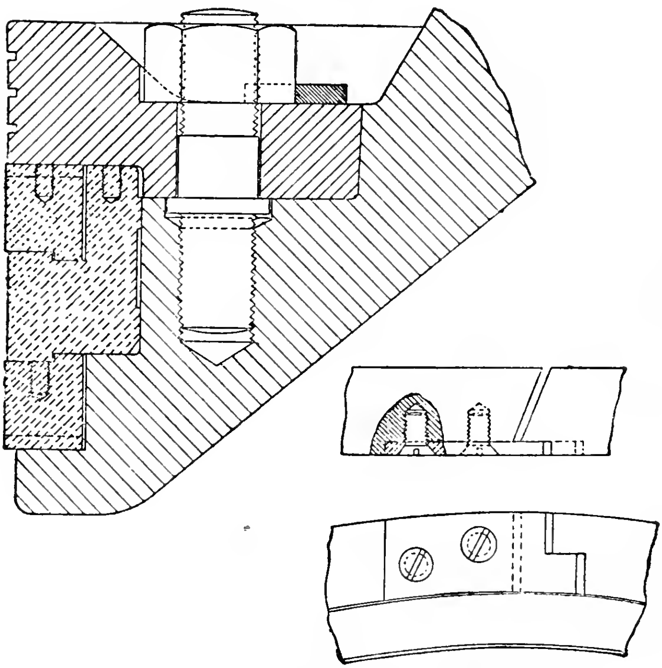


Fig. 88.—Restrained Packing-rings (Admiralty Plan).

Rowan's Patent and MacLaine's Patent each consists of two strong rings of square section, or of **U** section, pressed outwards by springs at the cross cut, and held against the flange and junk-ring by wave springs fitted in a groove between them. This arrangement has been found to work well.

Restrained Packings.—In the early days of piston-making designers were chiefly concerned in providing means for keeping the packing-rings on the cylinder walls; as pressures of steam increased they still experienced the same anxiety lest there should be loss by leakage. To-day we have less anxiety on that score, but a more serious, how to avoid the loss due to the rings being pressed unduly hard on the cylinder by the steam now used getting behind them. We cannot prevent the steam from getting behind the rings, so it is necessary to restrain them; this was first done by securing the ends or locking them together by the same device that set them. With the very high pressures now employed even that is not sufficient, so it is the practice in the Navy to fit solid restraining-rings, as shown in fig. 88, to keep the cut rings in place when fitted, as in right-hand illustration of same.

Body of the Piston.—Pistons of small size are usually made of a single thickness of metal without, of course, any stiffening webs or ribs (*v.* fig. 89).

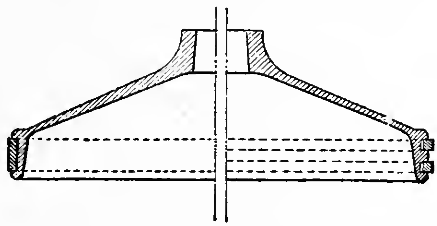


Fig. 89.—Forged-steel Pistons.

Pistons of very considerable size have been made of cast steel in this way; they are in the form of a cone, and shaped to suit the cylinder end (*v.* fig. 90); by that means they have the requisite degree of stiffness; originally they were made in this form to save weight, being for fast-running horizontal engines, but now they are used extensively in all fast-running engines. Cast-steel pistons are frequently used in the mercantile marine in engines of all sizes; chiefly, however, when in those running at high speeds, and always in those of large size. They may be used with advantage in most engines, and are not much more costly than hollow cast-iron ones, especially when of large

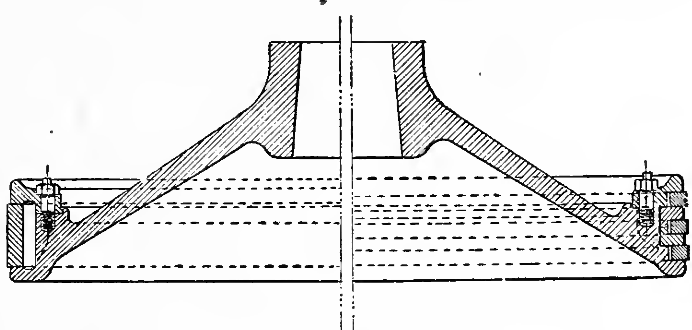


Fig. 90.—Cast-steel Pistons.

size. The pistons for very light, fast-running machinery are usually made of forged steel, and are very thin at the flange and near it. Solid cast-iron coned pistons can be used with advantage.

Pistons of ordinary marine engines above 20 inches diameter for H.P., and 40 inches diam. for L.P. cylinders, are usually made of cast-iron cellular —that is, with two thicknesses of metal stiffened or connected by ribs and webs, and either by the thickness of metal or by the depth of body made strong enough structurally to safely withstand, not only the load due to steam pressure exerted on it and transmitted to the rod, but also the shocks to which it is liable when priming occurs.

The piston body also must be so designed that it may be safely cast, for in the early days of large pistons it was not at all an uncommon thing for a piston to crack in cooling, or do so mysteriously afterwards. For this reason any rules must of necessity be empirical which set out the thickness of metal of the different parts of the body; but care must always be exercised by the designer that no one part is too small for the stresses to which it is subject. For example, there must be sufficient metal in the immediate neighbourhood of the piston-rod boss to resist the tendency to force out the centre by shearing the metal. Again, the piston may be taken as consisting of a number of sectors, and by considering one of such small sectors loaded with the pressure on its area at the centre of gravity of its figure, the bending moment at any section may be found, and the thickness of metal tried whether it be sufficient for the purpose.

For the section of an ordinary piston having a single rod, the following table gives the multipliers for obtaining the thickness of metal and sizes of the different parts.

Details of Construction of the Ordinary Piston.—Let D be the diameter of the piston in inches, p the maximum effective pressure per square inch on it, x a constant multiplier, found as follows:—

$$x = \frac{D}{50}(\sqrt{p} + 1).$$

For high-pressure cylinders p may be taken at half the absolute boiler pressure; for medium-pressure cylinders a quarter that pressure; and for low-pressure cylinders half the pressure, divided by ratio of low-pressure to high-pressure cylinders. For quadruple engines the value of p in the first medium-pressure is 0·3 the press., and in the second is 0·16 the absolute pressure.

Hollow Cast-Iron Pistons.

The thickness of front of piston near the boss	$= 0{\cdot}2 \times x$.
,, ,, ,, ,, rim	$= 0{\cdot}17 \times x$.
,, back ,,	$= 0{\cdot}18 \times x$.
,, boss around the rod	$= 0{\cdot}3 \times x$.
,, flange inside packing-ring	$= 0{\cdot}23 \times x$.
,, ,, at edge	$= 0{\cdot}25 \times x$.
,, packing-ring	$= 0{\cdot}15 \times x$.
,, junk-ring at edge	$= 0{\cdot}23 \times x$.
,, ,, inside packing-ring	$= 0{\cdot}21 \times x$.
,, ,, at bolt holes	$= 0{\cdot}35 \times x$.
,, metal around piston edge	$= 0{\cdot}25 \times x$.
The breadth of packing-ring	$= 0{\cdot}63 \times x$.
,, depth of piston at centre	$= 1{\cdot}4 \times x$.
,, lap of junk-ring on the piston	$= 0{\cdot}45 \times x$.
,, space between piston body and packing-ring	$= 0{\cdot}3 \times x$.
,, diameter of junk-ring bolts	$= 0{\cdot}1 \times x + 0{\cdot}25$ in.
,, pitch ,, ,,	= 10 diameters.
,, number of webs in the piston	$= \frac{D + 20}{12}$.
,, thickness ,, ,,	$= 0{\cdot}18 \times x$.

Solid Pistons.

	Cast Steel.	Cast Iron
Thickness near boss	$= 0{\cdot}26 \times x$	$= 0{\cdot}52 \times x$.
,, ,, rim	$= 0{\cdot}15 \times x$	$= 0{\cdot}22 \times x$.

Forged-Steel Pistons.

Thickness near boss	$= 0{\cdot}2 \times x$.
,, ,, rim	$= 0{\cdot}1 \times x$.

When made of exceptionally good metal, at least twice melted, the thicknesses of cast-iron pistons may be as much as 20 per cent. less than given by the rules; but, on the other hand, if made of other than really good metal, they should be thicker. The piston should be made of good metal always, and for fast-running engines it is better made of steel. The packing-ring was sometimes made thicker in the part opposite the cut than given above, in order to have sufficient elasticity of itself to press steam-tight against the cylinder; but it is better to let the springs perform their function wholly, and leave the ring to act only as the packing.

Junk-ring Bolts.—When screw-bolts are used to hold the junk-ring in place, they are either screwed into a brass nut let into a recess in the side of the piston (*v.* fig. 85*a*), or else screwed into a brass plug, which has been screwed tightly into the piston. The former plan is most general, and has the advantage that if the bolt thread is torn away the nut can be easily replaced, and owing to the length of body the bolts cannot slack themselves back. Some engineers, however, prefer to screw the bolts directly into the cast iron, making the tapped hole as deep as possible; and although it may be supposed the bolts would set fast by rust, practice has shown that such does not take place, nor do the cast-iron threads wear quickly away.

Studs are often used instead of screw-bolts (*v.* fig. 88), but although, to some extent, possessing advantages over the latter, they are not so convenient, and have all to be withdrawn when any refitting of the packing and junk-rings is necessary.

Safety-rings and Lock Bolts.—The vibration of the junk-ring has a tendency to slack back the bolts, and although it is a rare occurrence to find such a thing happen in a vertical engine, very serious accidents have frequently been caused by the junk-ring bolts getting loose in a horizontal cylinder. To prevent the possibility of a casualty the piston bolts of all engines should have a light wrought-iron ring secured to the junk-ring by studs, having square bodies and nuts secured with split-pins; this ring (*v.* fig. 88) fits close to the heads of the bolts, and prevents them then from turning. When studs are used their bodies should be square or heart-shaped with a projecting side, the holes in the junk-ring corresponding in size and form to them, so that, when on, it prevents them from unscrewing; the nuts may be prevented from slackening by a ring, or each stud may have a split-pin through its end.

There are some other methods, but none of them are either so efficient or so inexpensive as the above.

Solid Packings.—In order that the weight of the piston of a horizontal engine may be taken by the broad packing-ring, instead of by the comparatively narrow flange and junk-ring, it was customary and advisable to fit a cast-iron packing between the body of the piston and the packing-ring for about one-third of the circumference in lieu of springs. The pistons of diagonal and oscillating cylinders are also better if fitted in this way, and

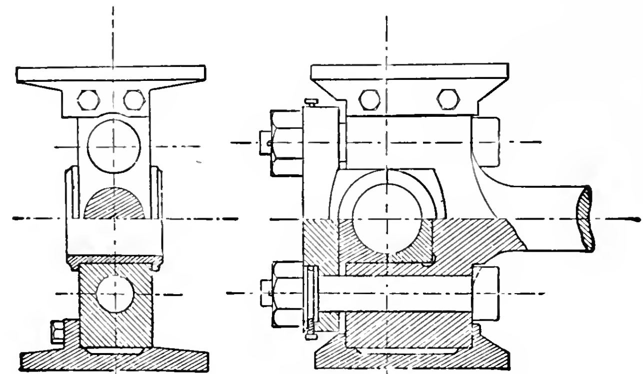


Fig. 91.

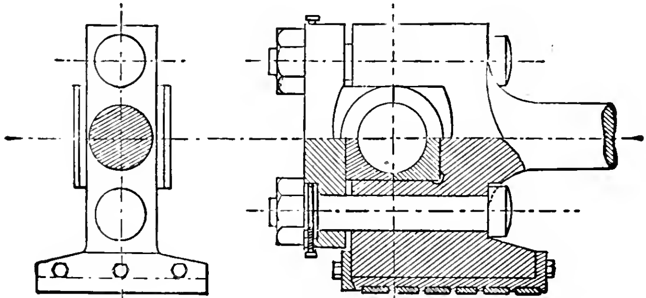


Fig. 92.

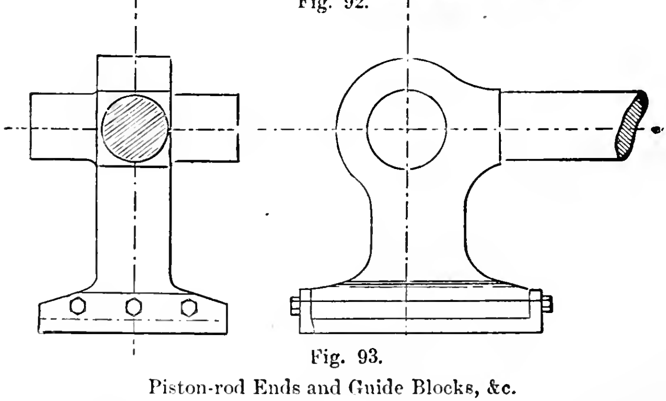


Fig. 93.

Piston-rod Ends and Guide Blocks, &c.

heavy pistons of vertical engines may be also dealt with in this way to provide for the heavy rolling of the ship.

Piston-rod.—It is usual to have only one rod to each piston of a direct-acting engine, but some manufacturers, to suit a particular style of crosshead and connecting-rod, fit two. The single rod is preferable from practical considerations, even for large engines, because it requires very considerable care on the part of the workman to bore the two holes in the piston, cylinder bottom, and crosshead so *exactly* that the rods will fit into their place without adjustment; the friction of the two stuffing-boxes will be very considerably more than that of the one larger one; the cost of labour will also be nearly double that for the single rod, and there are two stuffing-boxes, which require packing, and two glands demanding attention, instead of one.

Return connecting-rod and large steeple engines of necessity required two rods to each piston, and Messrs. Humphreys fitted four rods in the case of the very large pistons of H.M.S. "Monarch," the better to distribute the load over the piston face, and to admit of a better form of crosshead.

Diameter of Piston-rod.—Since the piston-rod is secured in the piston, and usually well guided at the other end, so that it cannot bend without meeting with considerable resistance, it may be treated as a *strut* or column, secured at both ends; but when the outer end fits into a crosshead, which would offer little or no resistance to bending, as in a paddle-wheel diagonal engine, then the rod must be treated as a column loose at one end and secured at the other.

From Mr. Hodgkinson's experiments, and Mr. L. Gordon's investigations the following are the formulæ for computing the strength of columns:—

(1) For a column fixed at both ends, $$P = \frac{f \, . \, S}{1 + a\dfrac{l^2}{d^2}}.$$

(2) For a column loose at both ends, $$P = \frac{f \, . \, S}{1 + 4\,a\dfrac{l^2}{d^2}}.$$

(3) For a column fixed at one end only, $$P = \frac{2\,f \, . \, S}{2 + 5\,a\dfrac{l^2}{d^2}}.$$

P is the load, l the length of the column in inches, d the diameter in inches, a for solid wrought iron and mild steel $\frac{1}{3000}$, and f 36,000 lbs. per square inch, S being the area of section of the rod in square inches. Taking this value of f in the above formulæ, P is the breaking load; since it is usual to have a factor of safety for all important parts of a marine engine, of at least 6, the value of f for this reason should not exceed 6,000 lbs. for mere strength; but as the piston-rod is liable to great shock, is always working with alternating stresses, and always receives its load suddenly, and must be rigid and without quiver, 3,000 lbs. should be taken as the value of f to calculate the diameter. These formulæ, however, are too complicated for general use, but the size of a piston-rod may be *checked* by them easily after having been calculated by an empirical formula.

Since, however, an approximate value may be safely taken for the relation between l and d of ordinary marine engines, the formulæ may be reduced to a very workable form. Hence, since

$$P = \frac{\pi D^2}{4} \times p; \qquad \text{and } f S = \frac{\pi d^2}{4} \times f.$$

D being the diameter of the piston, and p the effective pressure on it in pounds per square inch.

$$p \cdot D^2 = f d^2 \div 1 + a \frac{l^2}{d^2}.$$

Let

$$\frac{l}{d} \text{ be represented by } r,$$

Then

$$d = D \sqrt{\frac{p}{f} (1 + a r^2)}$$

taking $f = 3{,}025$.

Rule :—Diameter of piston-rod $= \frac{D}{55} \sqrt{p (1 + a r^2)}$. . . . (1)

The following are the values of r :—

Short stroke direct-acting engines, . . .	$r = 25 \div \sqrt{p}$
Long ,, ,, . . .	$r = 30 \div \sqrt{p}$
Very long ,, ,, . . .	$r = 35 \div \sqrt{p}$
Oscillating engines of short stroke . . .	$r = 40 \div \sqrt{p}$
,, ,, long ,, . . .	$r = 50 \div \sqrt{p}$
,, ,, medium ,, and compound,	$r = 37 \div \sqrt{p}$

The following rules, however, will give results sufficiently accurate for all practical purposes :—

Rule :—Diameter of piston-rod $= \frac{\text{diameter of cylinder}}{F} \sqrt{p}$.

The following are the values of F :—

Naval engines, direct-acting, ...	F = 50
,, return connecting-rod, 2 rods, . .	F = 70
Mercantile ordinary stroke, direct-acting, . . .	F = 45
,, long ,, ,, . . .	F = 42
,, medium ,, oscillating, . . .	F = 40

NOTE.—*Long and very long*, as compared with the stroke usual for the power of engine or size of cylinder.

D is the diameter of the low-pressure and d that of the high-pressure cylinder of a compound system; d_1, d_2, etc., the diameters of the intermediate cylinders; R is the ratio of low-pressure to high-pressure cylinder volumes; r_1 the ratio of volumes of the first intermediate and high-pressure cylinders; r_2 that of the second intermediate and high-pressure cylinders; if the stroke be the same in each cylinder

$$R = \frac{D^2}{d^2}; \; r_1 = \frac{d_1^2}{d^2}; \; r_2 = \frac{d_2^2}{d^2}.$$

The maximum pressure repeatedly applied on the pistons when working full speed are approximately as given by the following formulæ:—Where p_a is the absolute pressure at or near the engines, and may be taken as the load on safety-valve *plus* 15 lbs. for purposes of calculation of sizes of piston-rods, connecting-rods, columns, etc., and their bolts and fittings.

Values of p or *maximum* effective working pressure in

High-pressure cylinder of a compound engine, . . .	$p_a \times 0{\cdot}75$
Low-pressure ,, ,, ,, . . .	$p_a \div (R \times 1{\cdot}2)$
High-pressure ,, triple-compound engine, . .	$p_a \times 0{\cdot}63$
Medium-pressure ,, ,, ,, . .	$p_a \div (r_1 \times 1{\cdot}5)$
Low-pressure ,, ,, ,, . .	$p_a \div (R \times 1{\cdot}5)$
High-pressure ,, quadruple-compound engine, .	$p_a \times 0{\cdot}55$
1st medium-pressure cylinder of a quadruple-compound engine,	$p_a \div (r_1 \times 1{\cdot}65)$
2nd ,, ,, ,, ,, ,,	$p_a \div (r_2 \times 1{\cdot}55)$
Low-pressure ,, ,, ,, ,,	$p_a \div (R \times 1{\cdot}65)$

The piston-rods in a compound engine are all of one size, except in the case of the triple, with two low-pressure cylinders, whose rods may be and usually are smaller than those of the high-pressure and medium-pressure cylinders. It is at all times desirable that the piston-rod shall move through the stuffing-box without vibration, but especially is this so when metallic packing is used. It is, therefore, a good thing to have the piston-rods larger, rather than smaller, than given by the above rules.

Piston-rod Ends.—It is absolutely necessary that the rod should fit *perfectly* steam-tight into the piston, and also be of such a taper as not to "draw" in the least when subject to shock. If the rod end were made cylindrical or "parallel," as it is technically called, and fitted in to satisfy the above conditions, it would be very tedious and difficult to get it out again. For this reason principally it is usual to turn the part fitting steam-tight into the piston "taper" or conical. If the taper is very slight the rod can be easily made a tight fit, but unless formed with a shoulder at the end of the taper, it would in time become so tightly held by the piston as to withstand all attempts at withdrawal; moreover, there would be at all times a great danger of splitting the piston by the wedging action. If the taper extends the whole depth of the piston, it should be at the rate of ¾ inch to the foot; that is, the diameter of the rod at back is less than that at the front by one-sixteenth of the length of taper. Even with so liberal an allowance as this, great difficulty is often experienced in withdrawing the rod after a few months' work; for this reason, and to obtain a larger screwed end, some engineers do not extend the taper the full depth of the piston. The most convenient, and at the same time reliable, practice is to turn the piston-rod end with a shoulder of $\frac{1}{16}$ inch for small engines, and ⅛ inch for large ones, make the taper 3 inches to the foot (fig. 94) until the section of the rod is three-fourths of that of the body, then turn the remaining part parallel; the rod should then fit into the piston so as to leave ⅛ inch between it and the shoulder for large pistons, and $\frac{1}{16}$ inch when small. The shoulder prevents the rod from splitting the piston, and allows of the rod being turned true after long wear without encroaching on the taper.

It was usual to prolong the piston-rods of vertical engines to admit of the "tail" end passing through a stuffing-box in the cylinder-cover, and so help to guide the piston, and prevent its unduly wearing the cylinder. Since gravity prevents moisture getting to the packing of this stuffing-box, and the lubricant applied externally soon gets carried through to the cylinder, some trouble is experienced in keeping it steam-tight; the rolling of the ship also causes the piston to exert pressure sideways on the gland and packing, and further aggravates the evil; in other words, it is an unsatisfactory guide. For these reasons it is preferable to simply fit a brass or white metal bush in the cover for the "tail" end to work in, and case it with a dome or sheath fitted steam-tight and true on the cover; a couple of spiral grooves in the side of the bush will admit and release the steam. But, on the whole, it is very doubtful if these tail-rods are efficacious, and certainly they cannot be so beneficial to the good working of the cylinder as a piston with broad bearing surfaces. It should be noted that when the packing-ring is pressed out by springs acting independently of the body of the piston, it is advisable to form the piston with greater depth of flange and junk-ring.

The piston is secured to the rod by a nut, and the size of the rod at the

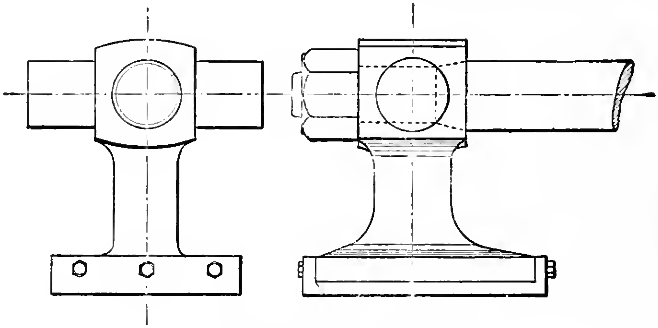


Fig. 94.—Piston-rod Crosshead.

nut should be such that the stress on the section at the bottom of the thread does not exceed 7,000 lbs. The depth of this nut need not exceed the diameter which would be determined by allowing this stress. To avoid the large cavity which is necessary in the cylinder-cover for the piston-rod nut, some engine builders recess it into the piston; this recess does not materially affect the strength of the piston, and the plan may be followed with advantage. Although piston-rod nuts seldom work loose, and those of vertical engines are less liable to this than are others, still as a measure of safety in all cases a taper-split pin should be fitted to the rod behind the nut, and in the case of large engines it is usual to fit a "lock" plate to the nut itself, or to adopt some other means of preventing it from moving at all when at work.

Cast-steel Piston-rod Crosshead.—Fig. 95 represents the modern form of crosshead made of cast steel and designed in such a way as a casting permits of. It is, of course, much lighter and cheaper than that of forged material, shown in fig. 94, and is especially suited to fast-running engines which require a large bearing surface. These crossheads can now be cast quite sound and of excellent material so as to be practically as good as a steel forging.

The stresses on the various sections are, as a rule, light, as the governing requirements of surface naturally causes it to be much larger than would be the case for mere strength. The gudgeons can be cast solid, or lightened out as shown. The example shown in fig. 95 has white metal fitted to both the head-going and stern-going sides, and is without a loose slipper. Such however, can always be fitted to this type of crosshead if desired.

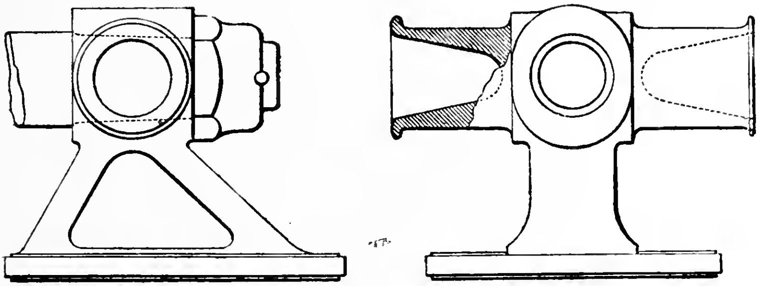


Fig. 95.—Cast-steel Crosshead.

When cast steel is objected to as material for the gudgeons, a composite design has been adopted whereby the crosshead is made of hard-wrought steel and fitted to the cast-steel slipper, etc., and secured by studs and nuts.

Piston-rod Guides.—The pressure on the piston is transmitted through the piston-rod to the connecting-rod, and the reaction of the latter rod acts in the direction of its length; consequently, when the connecting-rod is not in line with the piston-rod, the force of its reaction can be resolved into two component forces, one in the direction of the piston-rod, and the other perpendicular to it. This latter force is usually called the "thrust of the connecting-rod," and unless specially prevented, would tend to bend the piston-rod. To prevent such an occurrence, and to preserve the piston-rod in its true course, a guide is provided, and the piston-rod end fitted with blocks or slippers to work in it. This thrust varies from O at the end of the stroke to its maximum point, which is towards the point when the crank is

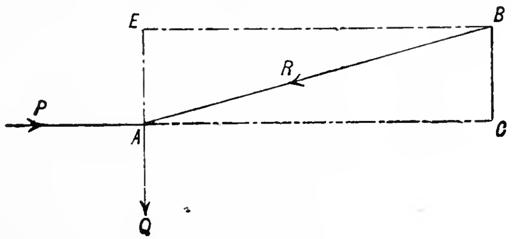


Fig. 96.

at a right angle to the centre line through the cylinder, and depending on the cut-off point; when steam is cut off past half-stroke, then, neglecting inertia effects, this is exactly the point of maximum thrust. To determine the magnitude of the thrust when P is the total effective load on the piston, S the stroke, and L the length of connecting-rod, represented by A B, fig. 96; completing the parallelogram by the dotted lines A E, B E, A C, then the reaction, R, of the connecting-rod, represented in direction and magnitude by A B, will be resolved into two forces, P and Q, represented by the lines E B, A E, both in direction and magnitude. P must equal the load on the piston, and Q the thrust on the guides, may be obtained by measuring B C on a graphically constructed diagram, or by geometry calculated as follows :—

$$AC^2 = AB^2 - BC^2 = L^2 - \left(\frac{S}{2}\right)^2$$

Now,

$$P : Q :: AC : BC.$$

Therefore,

$$Q = P \times \frac{BC}{AC} = P \times \frac{\frac{S}{2}}{\sqrt{L^2 - \left(\frac{S}{2}\right)^2}} = P \times \frac{S}{\sqrt{4\,L^2 - S^2}}.$$

Or, by Trigonometry,

$$Q = R \text{ sine } BAC,$$

$$P = R \text{ cosine } BAC, \text{ or } R = \frac{P}{\text{cosine } BAC} = P \text{ secant } BAC.$$

Therefore,

$$Q = P\frac{\text{sine } BAC}{\cos BAC} = P \tan BAC.$$

The angle B A C is found by knowing its sine to be half the stroke ÷ length of connecting-rod.

Example.—To find the thrust taken on the piston-rod guide of an engine whose piston load is 100,000 lbs.; the length of stroke is 60 inches, and the connecting-rod is 120 inches long.

$$\text{Thrust} = 100{,}000 \times \frac{60}{\sqrt{4 \times (120)^2 - 60^2}} = 25{,}819 \text{ lbs.}$$

Surface of Guide-block.—The area of the guide-block, or slipper-surface on which the thrust is taken, when going *ahead* should be sufficiently large to prevent the maximum pressure exceeding 100 lbs. per square inch. When the surfaces are kept well lubricated this allowance may be exceeded, but the reduction in surface should be effected by making shallow grooves and recesses in the face of the slipper, in which the lubricant can lodge and impart itself to the guide as it is carried along. A good method of carrying this into effect is to provide a surface calculated on the allowance of 100 lbs. per square inch, and by cross planing so as to leave shallow recesses about $\frac{1}{16}$ inch deep, reduce the actual surface which touches the guide to about $\frac{5}{8}$ of the original area; there will then be strips across the slipper $1\frac{1}{4}$ inches wide, with depressions between them $\frac{3}{4}$ inch wide, filled with grease.

When, however, the piston speeds exceed 300 feet per minute the area of guide-block should be increased to ensure continuous safe running, and follow this rule :—

$$\text{Working pressure per square inch} = 1{,}760 \div \sqrt{S + 100}.$$

For example, an engine running at a piston speed of 900 feet per minute should have guide-blocks such that the maximum pressure per square inch does not exceed $\frac{1,760}{\sqrt{900+100}}$, or 55 lbs.

Guide-blocks designed on these lines for *ahead* motion have always ample provision for *astern-going*. Then in any engine

$$\textit{Rule:}\text{—Gross area of guide-block shoe} = \frac{T \times \sqrt{(S+100)}}{1,760} \text{ sq. ins.}$$

Cast iron, hard and close-grained, is really good material for the guide plates; its surface, after a few days' work, becomes exceedingly hard and highly polished, and offers very little resistance to the slipper or guide-block. So long as this hard skin remains intact, no trouble will be experienced, but if abrasion takes place from heating or other cause, it rarely works well after and should be at once planed afresh, or, better still, ground smooth.

The slippers or facing plates fitted to the piston-rod or crosshead were sometimes made of bronze; but bronze never gets that smooth hard skin so essential to good and efficient working, and when once the surface is grooved and scratched, it will wear away very rapidly. Fine grain cast iron is, after all, the best metal for this purpose, if care is taken at the first working of the engine to run for a few hours at easy speed, so as to rub down and polish the surfaces; after this is once thoroughly done, cast-iron surfaces will continue to work well with very slight attention. White metal is, however, now generally used for the facing of slippers, and works very well, and for high speed is reliable for good and safe working. The best way of using white metal for this purpose is to fit strips of this material into grooves planed across a cast-iron slipper, and leave them standing from $\frac{1}{16}$ to $\frac{1}{8}$ inch above the cast iron. The strips should be about 2 inches wide, and the space between them from $\frac{3}{4}$ to $1\frac{1}{4}$ inches, into which the lubricant can collect and lodge, as before described. A slipper fitted in this way is shown in fig. 92. It is cheaper, however, and more general to cast the white metal practically over the whole surface of the slipper, where it is retained in place by "tinning" and underlying the edges of the recess provided for it. It is usual to cut oil-ways in the face of the guide, which distribute the oil across it, and metal combs secured to the slipper dip into the oil receiver at the end of the guide, and smear the face on the return stroke.

The guide plates are sometimes planed across, instead of the slippers, for the same purpose of retaining the lubricant, or have circular grooved rings dotted about their surface.

Piston-rod Crossheads and Gudgeons.—When there are two piston-rods, as in the case of the return connecting-rod engine, they are united to a common crosshead, having a turned journal in the middle for the connecting-rod to work on; or else a bearing is fitted to the crosshead, in which a gudgeon on the connecting-rod works. The former (fig. 97) is the better and usual plan under ordinary circumstances. This crosshead is of wrought-iron or steel, and made of a form suitable to the circumstances, and arranged to work in guides. The diameter at the middle must, of course, be sufficient to withstand the bending action, and generally from this cause ample surface is provided for good working; but in any case the area, calculated by multiplying the diameter of the journal by its length, should be such that the pressure does not exceed 1,200 lbs. per square inch. taking the maximum *load on the piston* as the total pressure on it.

Let L be the distance of the centres of the piston-rods in inches, and P the maximum load on the piston in pounds, then for strength

$$\text{Diameter of crosshead should not be less than } \frac{\sqrt[3]{P \times L}}{18}$$

For good wearing, l being the length of the journal.

$$\text{Diameter of crosshead should not be less than } \frac{P}{1{,}200 \times l}.$$

With fast-running engines the following rule may be followed where P is the maximum load on the piston, d is the diameter and l the total length of gudgeon or crosshead arms in inches, and R the revolutions per minute.

$$d \times l = \frac{P \times \sqrt{R + 100}}{12{,}500}.$$

Then for any engine

$$\text{Diameter of crosshead} = \frac{P \times \sqrt{R + 100}}{12{,}500 \times l}.$$

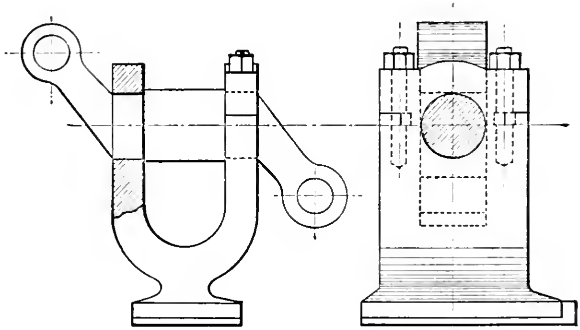


Fig. 97.—Crosshead and Guide-block for double Piston-rods.

Of course, the maximum load on the crosshead is really the reaction of the connecting-rod, but, to avoid any complication of the calculation, it is sufficient to take the load on the piston.

When the diameter of the crosshead journal is calculated by the first rule, the length is usually made equal to it.

Direct-acting engines have sometimes a gudgeon secured to the connecting-rod (fig. 98), which works in a bearing in the piston-rod end (figs. 91 and 92); or have the gudgeon at the piston-rod end (figs. 93, 94, and 95), and connecting-rod (figs. 99 and 100) swung on it by brasses, etc., on either side. By the latter plan larger bearing surfaces are obtainable, and the brasses, being on the outside of the rods, are much easier watched and adjusted; on the other hand, there are two sets of bolts, brasses, etc., to lubricate and keep in order, and there is the liability by careless adjustment to put the whole of the load on one side only. In the main, however, this plan is a preferable one for large and heavy rods, and it is one which admits of the piston-rod being fitted into its end (figs. 94 and 95) instead of forged with it. This latter advantage is well worthy of consideration, for it is highly important that the piston-rod shall be quite free from flaws and reeds on its

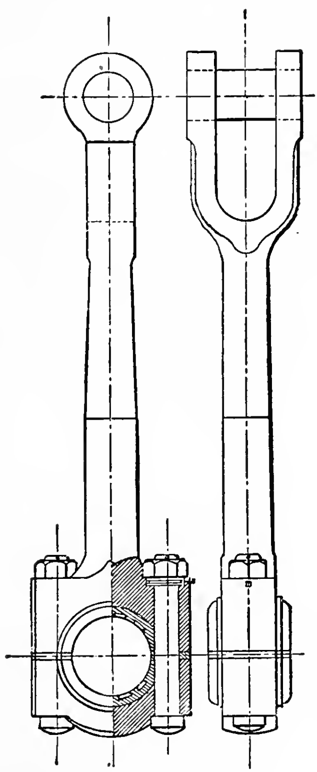


Fig. 98.

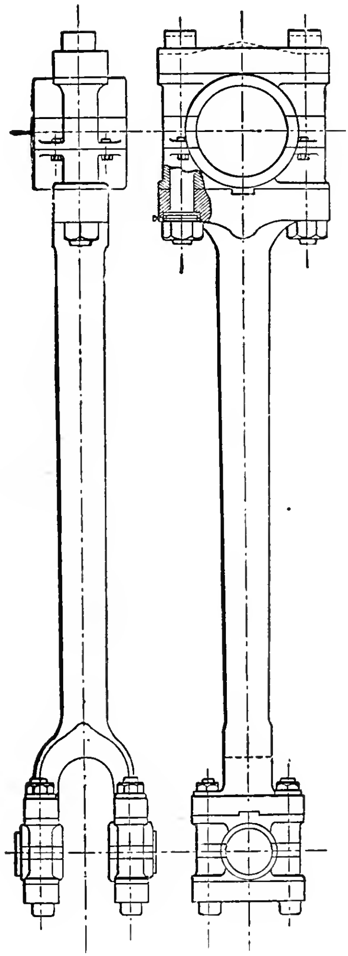


Fig. 99.—Connecting-rods.

surface, otherwise the packing soon gets damaged, and it is then found impossible to keep the glands from leaking. A steel rod rarely has them, but if the rod simply fits into the crosshead or rod end, this latter may with advantage be made of cast steel, while the rod may be of forged steel of any desired quality, easily returned or ground in the lathe and cheaply replaced.

Smaller engines are better with the gudgeon shrunk into the jaws of the connecting-rod, and working in brasses fitted into a recess in the piston-rod end (fig. 91), and secured by a wrought cap and two bolts.

The diameter of the gudgeon = 1·125 × diameter of piston-rod.
,, length ,, ,, = 1·4 × ,, ,,

The area obtained by multiplying these is exactly double the area of the piston-rod section, and so if the maximum stress per square inch on the rod does not exceed 2,400 lbs., the above rule holds good; if this allowance of stress is exceeded in calculating the rod, then the *length* of the gudgeon should be increased until the pressure on the section, as calculated by multiplying length by diameter is in accordance with the following:—

Rule.—Pressure per square inch on gudgeon $= \frac{12,500}{\sqrt{R} + 100}$ lbs. for revolutions R per minute.

When the gudgeon is fixed in the piston-rod (fig. 93), or formed with the guide ends as in fig. 94, the length *of each* end should be not less than 0·75 the diameter of the piston-rod.

The brasses when fitted into the piston-rod should be square-bottomed and of hard gun-metal, with good oil-ways, as owing to the motion being through a comparatively small angle only, the lubricant is not so easily spread. Some white metal *does not* work well on gudgeons and crossheads; it has a tendency to abrade, and to *wear the journal* oval. This is generally observed whenever white metal is used on a bearing subject only to a small angular motion. White metal, however, can be used with mild steel gudgeons, especially in fast-running engines with advantage.

Gudgeons, when fitted to the connecting-rods, should be made of hard steel or mild steel case-hardened; great care is required, if of the latter, that they are carefully ground true after hardening.

The bolts securing the brasses should be of the best tempered mild steel; they should be of such a size that the stress per square inch of section *at the bottom of the thread* does not exceed 6,500. It is true that where extreme lightness of machinery is a *sine quâ non*, these stresses have been exceeded by some engineers; but even under these circumstances it is unwise to exceed them by more than 10 per cent., as the saving of weight effected is exceedingly small, especially when compared with the risk run (*v.* Table lxxix.).

When the bolts are less than 2 inches in diameter, owing to the uncertainty of the depth of thread, etc., the allowance should be not more than 6,000 lbs.

Rule.—Diameter of piston-rod bolts at the bottom of thread

$$= D\sqrt{\frac{p}{f}}.$$

D is the diameter of piston, p, the effective pressure on it per square inch.

Where there are two bolts of steel f = 13,000 lbs.; under 2″, 12,000.
,, four ,, f = 24,500 lbs.; under 2″, 22,000.

The cap is of the same width as the piston-rod end—viz., 1·15 × diameter of the rod, and its thickness equal to the diameter of the bolts at *the bottom of the thread.*

Connecting-rods.—The length of the connecting-rod measured from the centre of the gudgeon to the centre of the crank-pin should, if possible, be not less than twice the stroke. Quick-running vertical engines have almost invariably the connecting-rods twice their stroke in length; other vertical engines from twice to three times, but generally two and a quarter times is not exceeded.

A connecting-rod may be viewed as a strut loose at both ends, the formula for which, as given on p. 271, is $R = \dfrac{f\,S}{1 + 4\,a\,\dfrac{l^2}{d^2}}$.

The value of R will be found by multiplying the load on the piston by the secant of the angle of obliquity of the connecting-rod (*vide* p. 276). Or by geometry

$$R = P \times \frac{2\ L}{\sqrt{4\ L^2 - S^2}},$$

P being the load on the piston, S the stroke, and L the length of the connecting-rod as before.

Simplifying the above formula by assuming a value r for the ratio of l to d, and substituting $\frac{\pi\ d^2}{4}$ for S; and taking the value of f at 3000 lbs.

$$\text{Diameter of connecting-rod in the middle} = \frac{\sqrt{R\ (1 + 4\ a\ r^2)}}{48 \cdot 5}.$$

Example.—To find the diameter at the middle of the connecting-rod of an engine, 60 inches stroke, whose length is 120 inches, and the load on the piston 100,000 lbs., r being taken at 15.

$$R = \frac{100{,}000 \times 2 \times 120}{\sqrt{4 \times 120^2 - 60^2}} = 103{,}358 \text{ lbs.}$$

$$\text{Diameter at middle} = \frac{\sqrt{103{,}358\left(1 + \dfrac{4 \times 15^2}{3000}\right)}}{48 \cdot 5} = 7 \cdot 6 \text{ inches.}$$

The following are the values of r in practice :—

Naval engines—Direct-acting		. .	$r =$ 9 to 11.
Mercantile engines,	,,	ordinary .	$r =$ 12.
,,	,,	long stroke	$r =$ 13 to 16.

Taking 10 as the average value of r for naval engines, and 13 for mercantile, then,

For a naval engine,

$$\text{Diameter of connecting-rod at middle} = \sqrt{\frac{R}{2000}}.$$

For mercantile engines,

$$\text{Diameter of connecting-rod at middle} = \sqrt{\frac{R}{1900}}.$$

The sizes given by these rules, although large enough for strength, are somewhat smaller than found in actual practice in the mercantile marine generally.

The following empirical formula will be found a very useful one, and the results given by it agree very closely with good modern practice :—

$$\text{Diameter of connecting-rod at middle} = \frac{\sqrt{L \times K}}{4}.$$

L is the length of the rod in inches, and

$$K = 0\cdot03 \times \sqrt{\text{Effective load on the piston in lbs.}}$$

Example.—To find the diameter of the connecting-rod, 100 inches long, for an engine having a load of 55,000 lbs.

$$K = 0\cdot03 \times \sqrt{55{,}000} = 7\cdot0,$$

$$\text{Diameter} = \frac{\sqrt{100 \times 7}}{4} = 6\cdot6 \text{ inches.}$$

The diameter of the connecting-rod at the ends may be 0·875 of its diameter in the middle. The tapering of rods, or making them barrel-shaped, is usual in the case of those having single brasses at both ends; then the diameter of the crank-pin end is 0·925 of the diameter at middle. Direct-acting engines have usually the connecting-rods tapering from the gudgeon end to the middle, and then parallel, or nearly so, to the crank-pin end.

It is, however, simpler and sufficiently accurate to follow, in the case of connecting-rods, a similar rule to that laid down on p. 272 for piston-rods, viz. :—

$$\text{Diameter of connecting-rod} = \frac{\text{Diameter of cylinder}}{F} \sqrt{p}.$$

Here p is as calculated by the formulæ on p. 273. F may be taken at 55 for the crosshead end of fast-running light engines, and 52 of mercantile engines of ordinary type. The diameter of the rods at the middle may be got by dividing by F — L, where L is the length of connecting-rod in feet.

Connecting-rod Bolts.—The diameter of the bolts may be calculated by allowing the same stress per square inch as that given for piston-rod bolts. It is usual now, from practical considerations, to make the bolts of both piston and connecting-rod of the same size: the bolts therefore should be calculated from the load on the connecting-rod. In order that the whole of the stretch shall not come on one section, as at the bottom of the last thread of an ordinary bolt, it is better to turn part of the body of connecting- and piston-rod bolts to the same diameter as at the bottom of the thread, leaving it a little larger than the diameter over the thread close to the head, and in way of any joint—that is, the bolt is made with a *plus* thread, and bearing collars where required.

Connecting-rod Brasses.—The crank-pin brasses are more severely tried than any others about an engine, and, therefore, should be most carefully designed, and made of the very best material. Some engineers make the brasses to form the end of the rod (fig. 99), and retained to it by bolts and a steel cap: others prefer that they shall only act as bushes or liners to the connecting-rod, sometimes fitting them into a square or octagonal recess in the rod end, and held in place by a flat cap and bolts, just as is generally done to piston-rod ends; but more generally they are fitted in duplicate halves, as shown in fig. 98. The former plan is an expensive one when they are of large size and made of brass, on account of the great weight required, and consequently are also costly to renew when worn, besides which they are very liable to get out of shape when heated, and to crack through the crowns. The latter plan avoids the use of so much brass, gives a good solid bed to the brasses, and leaves the bolts free of all stress except tension. When rods are made in this way it is usual to forge the head of the rod solid, and turn it and the cap at the same operation; the hole for the brasses is bored or slotted out (the latter when the hole is 9 inches and upwards in diameter) roughly; the head is then slotted through or parted in the lathe so as to cut off the cap, the space left by the tool being equal to twice the difference in thickness of the brass at the crown and sides; the cap is then bolted close to the rod, and the hole bored out to the diameter of the brasses measured across the rod. The brasses are kept from turning by a brass distance piece secured between the cap and rod and projecting between the brasses, and in the case of large brasses a short feather is fitted close to each flange in the crown. All brasses have a tendency to close on the pin or journal after having been hot, because the inner surface becomes warm first, and the metal in expanding tends to straighten the curved part; this is resisted by the other part of the brass and the bed in which it is fitted, and in consequence this inner surface gets compressed permanently, so that on cooling down it contracts, and tries then to give the brass more curvature, and so presses hard on the journal. It is for this reason that some bearings will never work cool but always a trifle warm; this slight amount of heat causes the brass to expand so as truly to fit the journal. It is now not at all uncommon to make these fittings of good cast or malleable cast iron when lined with white metal; indeed, cast iron carries white metal better than brass does, and when of really good mixture is quite as strong as gun-metal, and stronger than the common brass so often used with white metal.

Fig. 100 is an example of a modern connecting-rod with the jaws made as part of a cone, instead of as in fig. 99, thereby providing a better connection between each jaw and the body of the rod, as well as permitting of its being machined much more cheaply. This rod is fitted with cast-iron bearings at each end, lined with white metal. This form of rod and "brasses" is now very generally used in the mercantile marine and Navy for engines of all sizes.

White metal is better than bronze for the rubbing surface of the crank-pin "brasses," it is important, therefore, that the white metal shall project beyond the "brass," so that it alone shall bear on the pin. For this purpose strips of white metal should be fitted into grooves planed in the "brass" and be well hammered, so as to thoroughly fill the spaces, after which it should be smoothly bored and fitted to the pin. Brasses which have not been originally designed for white metal may be fitted in this way, or by boring some shallow holes, whose diameter at the bottom is more than at the surface, casting into them buttons of white metal, which, after hammering down, are bored out so that the white metal stands out beyond the original wearing surface.

A very good plan, but somewhat more expensive and not more efficient than the one above described, is to run the white metal into recesses cast with the brass, hammer it well in place, bore out, and then plane out the brass intervening between the white metal patches, leaving only slight ridges surrounding the latter, which prevent it from being spread out. The most

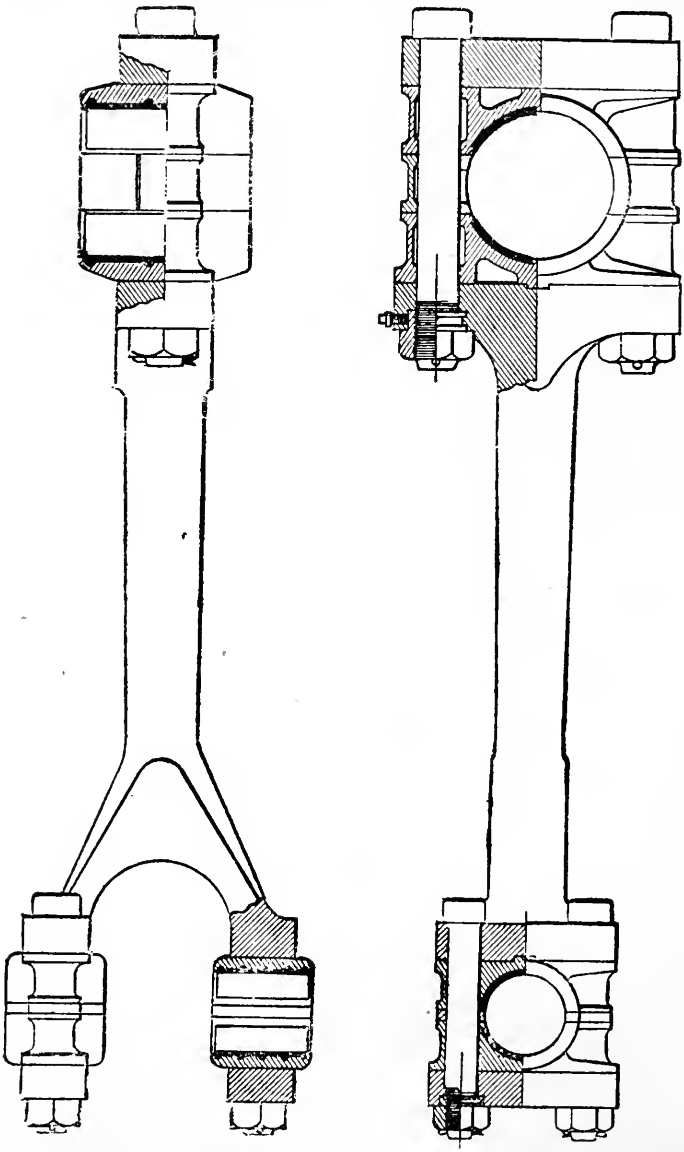


Fig. 100.—Connecting-rod. Modern Form.

general plan, however, is to have a recess in the "brass" or shell, which is now often of cast iron instead of bronze, tin the surface and run the metal in place while the shell is warm. When cold bore out as usual, trim the edges, etc., and cut grooves for the oil to circulate in the surface of the metal (*v.* fig. 100).

Caps of Connecting-rod Brasses.—The width of the connecting-rod end should be such as to efficiently support the brasses; its thickness (in direction of the length) = 0·6 × diameter at middle of rod.

The thickness of cap at middle = 0·8 × diameter of body of bolts + 0·1 × pitch of bolts.

Thickness of cap at ends = diameter of bolts at bottom of thread.

For ease of manufacture caps are generally made straight and of thickness given by the first rule.

Gudgeon End of Rod.—The jaws of every connecting-rod are subject to forces which tend to open them on the down stroke and close them on the up. When fitted with a gudgeon-pin secured in the eyes, as in fig. 98, the jaws are subject to bending moments at the various sections, which become maxima at a point just below the gudgeon and also at an angle of about 40° to the vertical through the centre of curvature of the inner portion. When the rod is fitted with "brasses" working on a crosshead, as in fig. 99, the maximum bending moment is at an angle of about 40° to the vertical or axis of the rod, and is greater than when with a gudgeon secured to the rod ends. When the diameter of the gudgeon or crosshead ends is

$$= \frac{\text{Diameter of cylinder}}{45} \times \sqrt{p},$$

then

Let P be the maximum re-current load on the piston.

L the width from centre to centre of gudgeon-rings or of crosshead brasses when so fitted.

$$F = \sqrt{\frac{P}{3{,}000}}, \text{ then}$$

Diameter of gudgeon ring .	. = diameter of gudgeon + 1·25 × F.
Thickness ,, ,, .	. = 0·85 × F.
Width of jaw . . .	. = 1·35 × F, or 1·1 × diameter of rod.
Thickness of jaw in line at 40°	. = ·48 F × L when with gudgeon.
,, ,, ,,	. = ·52 F × L when with brasses.

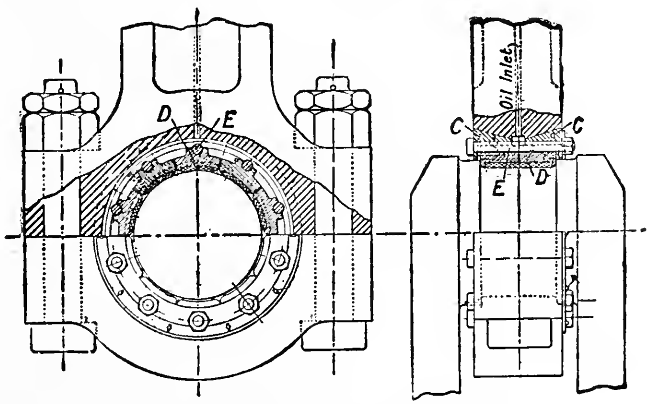


Fig. 100*a*.—Connecting-rod End (Michel).

CHAPTER XII.

SHAFTING—CRANKS AND CRANK-SHAFTS, ETC.

The Shafting of a Modern Marine Engine is made wholly of mild steel, produced from a Siemens furnace, and generally having a tensile ultimate strength of about 30 tons. In section it is, of course, circular, and in the mercantile marine generally solid; in Naval ships it is always made hollow by boring out the centre so as to remove the core of doubtful metal and provide the strongest shaft with the least weight, inasmuch as, with a large reduction in material, there is only a small reduction in strength by this treatment. For example, if a shaft have a borehole in diameter half that of the outside, the reduction in weight is 25 per cent., while the strength is diminished by $12\frac{1}{2}$ per cent. only. The cost of hollow shafts is, of course, greatly in excess of that of solid ones; notwithstanding, it is found worth while to fit them to many of the high-speed express steamers crossing channels and entering shallow harbours.

In Practice every Shaft in a Ship is subject to torsion, and, therefore, to the shearing stresses caused by torque. It has also to resist the additional shearing stress due to its own weight, but this addition is, as a rule, not great by comparison. But the weight causes a bending moment to act on every part of the shaft, which, if the bearings are wide apart, will cause a stress and deflection which must be taken account of.

Throughout the Line of Shafting from the motor to the propeller is transmitted to the latter the torque generated in the former, and when the screw is heavy and racing badly in a sea-way, there may be, and often is, a reverse action by the inertia stored in the latter transferring back a torque to the motor.

The Propeller-shaft of every Ship is subject to a more severe load from the overhung heavy instrument setting up severe bending moments, due to gravity in smooth water, and in rough water to the inertia effects and the uneven action of the blades on the water. When a ship is pitching in a heavy sea the stern drops with great rapidity, so that the vertical velocity of a heavy screw is serious; this velocity is checked with sufficient abruptness to put a very heavy bending moment and shearing force on the shaft end, where it ceases to be supported by the stern bush, as is equally the case when the wave motion causes a rapid rise of the stern. The side throw or lurch at the stern from wave action sets up horizontal inertia forces, also severe and trying, although not so intense as the vertical ones. The differential pressures on opposite blades of the screw also produce bending moments with the corresponding shearing forces. Hence the stern shaft of a cargo steamer, which is sometimes deeply laden and sometimes light, and always somewhat lively at the stern, is tried very severely, and should always be of ample size;

in fact, larger than that of the finer-lined express steamer of the same power, but with a screw lighter, because of smaller diameter and often of bronze. Crank-shafts of reciprocators are subject to complex loads at their various parts, and always have bending moments of kinds as well as torque to resist. **The shafts of turbines** also have to bear considerable bending moments, due to the weight of the rotor, but in their case both it and the torque are constant and uniform; there is no variation in torque through the revolution as with the reciprocator. It does, however, have to resist inertia forces of a kind when the ship is in a sea-way, and when the turbine is well aft, as in fig. 40, the stresses due to them will be heavy in bad weather.

The Crank-shaft of the ordinary Oil Engine is subject to severe bending moments, and to shocks which produce heavy shearing forces. It is claimed for the Diesel engine that, owing to the very great amount of compression, there is little or no shock at explosion, and the diagram (fig. 101) would seem to prove this. The combined torque diagram from an oil engine on the

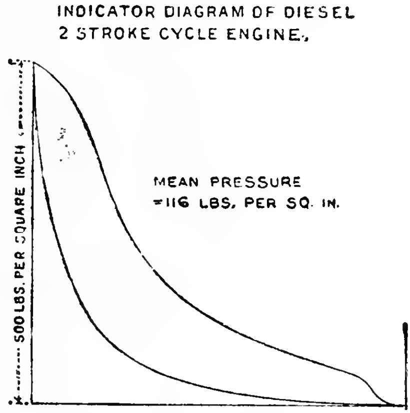


Fig. 101.—Diesel Engine Diagram.

four-cycle system, however, shows a very large ratio of maximum to mean torque as well as the big bending moment.

The Tunnel or Intermediate Shafting of a ship is subject theoretically to torque only; it is, of course, in compression axially from the screw shaft to the thrust block, but this affects the torque so little as to be negligible (about $1\frac{1}{2}$ per cent.). If the tunnel bearings are close together, and the rate of revolution is not high, then torsion alone is the governing factor in determining its size. But, since the rate of revolution has increased so largely, especially where turbines are employed, it is necessary now to consider the bending moments that may or do come on these shafts. However close, in reason, the bearings may be to one another, the weight of the shaft must cause a certain amount of sag; if the centre of gravity of the shaft does not coincide with its axis of rotation, it may then whirl—that is, the C.G. will tend to move in a circle, whose centre is on the axis of rotation. At low speeds the whirling action is comparatively small, and not sufficient to further bend the shaft, but at higher speeds the centrifugal force may be so great as to be harmful.

In order that a shaft may revolve freely without any danger of whirling due to the "sag" or deflection caused by its own weight, the bearings should never be further apart than the distance given by the following rule, where d is the diameter in feet and z is a multiplier depending on the rate of revolution :—

Rule.—Pitch of plummer blocks should not exceed $z\sqrt{d}$.

When the revolutions	do not	exceed	200	per minute.	$z = 40.$
,,	,,	,,	250	,,	$z = 36.$
,,	,,	,,	300	,,	$z = 31{\cdot}5.$
,,	,,	,,	400	,,	$z = 29{\cdot}0.$
,,	,,	,,	500	,,	$z = 25{\cdot}5.$
,,	,,	,,	600	,,	$z = 23{\cdot}5.$
,,	,,	,,	700	,,	$z = 21{\cdot}5.$
,,	,,	,,	800	,,	$z = 20{\cdot}0.$
,,	,,	,,	900	,,	$z = 19{\cdot}0.$
,,	,,	,,	1,000	,,	$z = 18{\cdot}0.$

Considering that in a screw ship the lengths of tunnel shafts near the stern will be subject to considerable inertia forces, the pitch of bearings should be less than the limit given above for still water, and about 60 per cent. should not be exceeded. For example, a shaft 12 inches diameter at 200 revolutions should be supported every 24 feet, while at 400 revolutions it should be 17·4 feet, and never exceed in smooth water 29 feet.

Alternating Stresses.—If a shaft having a heavy body, such as a screw or paddle-wheel, at its overhung end is revolving under the action of torque, the material from the weight to the supporting bearing is subject to tension and compression, and shearing due to that weight and to shearing at every section throughout its length due to the torque. But the bending moment due to the overhung weight sets up tension on all particles above the neutral axis and compression on all below it, and inasmuch as the shaft is turning, so that what was at the top at one moment is at the bottom the next, hence the particles, from being subject to tearing apart when above, are quickly in a state of being crushed together below. The magnitude of the forces ranges from 0 steadily to the maximum at top and bottom, so that the load and change of load is gradual; nevertheless, seeing that at 120 revolutions the change from plus to minus is no less than four times in a second, while with turbine-driven shafts there is often as many as 24 alternations per second, the endurance of the material is severely tried. Such stresses as these which it has to resist are called *Alternating*.

Piston-rods, Connecting-rods, Valve-rods, and Columns are all subject to these reversals of load and stress, but in their case the application and release are sudden and made with more or less shock, but modified somewhat by the action of inertia forces and the effect of cushioning.

The Arm of a Lever vibrating through an angle when transmitting energy is subject to alternating stresses also, more or less abruptly applied and released.

But the Arm of a Crank-shaft, which is a lever moving always angularly in one direction, is subject to no such changes of stress. On the one side on which the force is applied the molecules are subject to tension only, and those on the opposite side to compression only; but in the case of an engine crank the load and stress, as measured by the turning moment, are not so abrupt in their application, etc., and they also are modified by inertia forces and cushioning. With the reversal of the engine into "astern gear" these stresses are, of course, reversed.

The Bolts of Connecting-rods, Main Bearings, Cylinder Feet, and Cylinder Covers are all subject to a load of one kind, but it is applied and released, then ceases for a moment, to be again applied more or less suddenly. Such loads and the stresses caused by them are called *Intermittent*.

Iron, Steel, and all Materials suffer more or less severely from these intermittent and alternating stresses, but much more so from the latter, as the effect on their structures is to gradually destroy them. The higher the stress and the greater the number of alternations or intermissions per minute, the quicker is the strength or virtue of the material annihilated. Of steels some varieties have a longer life than others, and of the bronzes it is even more marked. It does not follow of necessity that because a steel or bronze has a very high elastic limit that it can be used with greater longevity in a rapidly active structure than one whose elastic limit is less. Professor Arnold has shown that bar steel, by rolling and drawing cold, might have its yield point raised by as much as 8 to 12 tons, while the endurance was reduced by 38 per cent.; in the case of aluminium copper alloys the endurance of one was more than double that of others not differing largely from it in composition.

The Influence of Numbers of Reversals of Stress and their magnitudes has been demonstrated also by Prof. Arnold and others, and the results of experiments on steel and iron is startling.

Wöhler's experiments showed that the iron used for axles then (1870) required 56,430 repetitions with a plus and minus load of 15·3 tons per square inch to produce fracture, while with 8·6 tons each it required over 19 millions. Similar experiments with Krupp's steel showed that with plus and minus loads of 20·1 tons per square inch 55,100 repetitions produced fracture at 15·3 tons; it was done with a little over 3 millions, while some bars with 14·3 tons took over 45 millions. It is clear, then, that since some definite number of alternations will break down the tenacity of a metal, and that the greater the stress the fewer they will be, that for long life a low working stress is necessary.

Safe Working Stress on the material must, therefore, vary with the conditions under which it performs its duty. In a general way the highest stress allowable under any circumstances is one-half that of the elastic limit, and for margins of safety to cover inaccuracies and small hidden defects the highest working stress should not exceed 40 per cent. of the elastic limit. If the material is subject to intermittent stress, 90 per cent. of this maximum or 36 per cent. of the elastic limit should be observed in the design. If, however, the stresses are alternating, 66 of the maximum or 26·4 per cent. of the elastic limit should not be exceeded.

The elastic limit of mild steel, as found in any workshop, is about

30,000 lbs. The working stresses on it with moderate speed of revolution should be 12,000, 10,800, and 7,920 lbs. per square inch in tension.

For high rate of revolution even these allowances are too great, and over 500 revolutions per minute the reduction in stress should be quite 10 per cent., so that for screw shafts of fast-running engines the stress allowed in calculating the sizes should not exceed 7,000 lbs. for continuous working.

For work where the alternations or intermissions are effected with considerable shock the working stress should be at least 10 per cent. less than when effected gently.

It should be noted, however, that certain qualities of steel and bronze stand shock and alternation of stress much better than others.

Twisting Moment.—If a force is acting on a shaft so as to turn it, or tend to turn it, round on its axis, it is called a *twisting force*, and the *effort* of this force is measured by multiplying it by its distance from the axis, and called the *twisting moment* or *torque.* Suppose P is the thrust along the connecting-rod when at right angles to the crank, and L is the distance of the centre of the crank-pin from the centre of the shaft, P × L is the twisting moment on the shaft.

When *one* force is acting on the shaft as above described, the second force, which completes the *couple*, is the reaction of the bearing, which is equal to P, but acts in the opposite direction. If the force P and the reaction R act in a plane *perpendicular to the axis* of the shaft, they will cause no bending action on the shaft, but there will be a force R tending to shear the shaft across. But in actual practice it is almost impossible that P and R shall act in such a plane, and they usually act in planes parallel to one another, and perpendicular to the axis; hence, the shaft is also subject to a bending action. But if a shaft is turned by means of *two equal* forces acting in opposite directions, one on either side of the shaft and equidistant from the axis and in the same plane, then the shaft is balanced, these forces will cause no pressure on the bearings, and it is subject, therefore, to twisting strains only. If one shaft is coupled to another shaft, from which it is to transmit power by two coupling bolts equidistant from the centre, it will only receive a twisting strain. Such is the state of the shafting from the crank-shaft to the propeller-shaft of a screw steamship.

Resistance to Twisting.—Let T be the twisting moment on a shaft in inch pounds, d the diameter of the shaft in inches, and f the stress per square inch on the transverse section of the shaft. Then (Rankine, *Applied Mechanics*, p. 355) for solid shafts of diameter d,

$$T = \frac{\pi d^3}{16} f = 0{\cdot}1964 f d^3; \text{ or } d = \sqrt[3]{\frac{T}{f}} \times 5{\cdot}1.$$

For hollow shafts with a bore diameter d_1,

$$T = 0{\cdot}1964 f \left(\frac{d^4 - d_1^4}{d}\right).$$

Example.—To find the diameter of a shaft subject to twisting only, the force being 100,000 lbs. acting at a distance of 24 inches, stress to be 8,000 lbs.

$$T = 100,000 \times 24 = 2,400,000 \text{ inch-lbs.}$$

$$d = \sqrt[3]{\frac{2,400,000}{8000}} \times 5{\cdot}1 = 11{\cdot}5 \text{ inches.}$$

Diameter of a Shaft subject to Torsion.—If a constant force P were applied to the crank-pin tangentially to its path, then the work done per revolution will be $P \times \frac{2\pi L}{12}$; L being the length of the crank in inches; then if R be the number of revolutions per minute,

$$\text{Work done per minute} = P \times \frac{2\pi L}{12} \times R. \quad (1)$$

But this work is equal to I.H.P. × 33,000; and the twisting moment is P × L constantly. Then

$$(P \times L) \times \frac{2\pi}{12} \times R = \text{I.H.P.} \times 33,000,$$

and

$$P \times L = \frac{\text{I.H.P.} \times 33,000 \times 12}{2\pi \times R}.$$

That is,

(2)

3)

$$\text{Diameter of shaft} = \sqrt[3]{\frac{\text{I.H.P.}}{R}} \times 42{\cdot}84. \quad (4)$$

But as shafts must be strong enough to resist the *maximum* twisting stress, it is necessary always to base calculations on it instead of on the mean twisting moment. The factor 42·84 must, therefore, be multiplied by the ratio of maximum to mean moment, as given in Table xxxv.

Professor Rankine directs (*Rules and Tables*, p. 250), in order to find the greatest twisting moment from the mean: if a shaft is driven by a single engine, multiply by 1·6; if by a pair of engines with cranks at right angles, by 1·1; if by three engines with cranks at angles of one-third of a revolution, by 1·05.

These values are, however, very much lower than usually obtained in modern practice if the effect of inertia forces are neglected.

For a three-crank engine, cranks at 120°, cutting off at half to two-thirds stroke, multiply by 1·15.

For a two- and four-crank engine having cranks at right angles, cutting off steam at half to three-quarter stroke, multiply by 1·3.

For a single-cylinder engine cutting off steam at half stroke, by 2·0.

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The reference on page 291 to Table XXXV. should be

Table XXXIV., page 298.

transmit power by two coupling bolts equidistant from the centre, only receive a twisting strain. Such is the state of the shafting from the crank-shaft to the propeller-shaft of a screw steamship.

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and

$$P \times L = \frac{\text{I.H.P.} \times 33,000 \times 12}{2\pi \times R}.$$

That is,

$$\text{Mean twisting moment} = \frac{\text{I.H.P.}}{R} \times 63,000. \quad - \quad - \quad - \quad - \quad (2)$$

And as before

$$d = \sqrt[3]{\frac{\text{I.H.P.} \times 63,000 \times 5{\cdot}1}{R \times f}}$$

$$= \sqrt[3]{\frac{\text{I.H.P.}}{R} \times \frac{321,300}{f}}. \quad - \quad - \quad - \quad - \quad (3)$$

If f be taken at 7,500 for mild steel

$$\text{Diameter of shaft} = \sqrt[3]{\frac{\text{I.H.P.}}{R}} \times 42{\cdot}84. \quad - \quad - \quad - \quad - \quad (4)$$

But as shafts must be strong enough to resist the *maximum* twisting stress, it is necessary always to base calculations on it instead of on the mean twisting moment. The factor 42·84 must, therefore, be multiplied by the ratio of maximum to mean moment, as given in Table xxxv.

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These values are, however, very much lower than usually obtained in modern practice if the effect of inertia forces are neglected.

For a three-crank engine, cranks at 120°, cutting off at half to two-thirds stroke, multiply by 1·15.

For a two- and four-crank engine having cranks at right angles, cutting off steam at half to three-quarter stroke, multiply by 1·3.

For a single-cylinder engine cutting off steam at half stroke, by 2·0.

The following rule holds good for the ordinary engines, as found in general use in the merchant service :—

$$\text{Diameter of the tunnel shafts} = \sqrt[3]{\frac{\text{I.H.P.}}{\text{R}}} \times \text{F.} \quad \cdot \quad \cdot \quad (5)$$

Two-stage compound engines, cranks at right angles—

$$\text{F} = 8\sqrt{p},$$

where p is the absolute actual pressure, or boiler pressure + 15 lbs.

Triple-compound, three-cranks at 120°—

$$\text{F} = 4{\cdot}8\sqrt{p}.$$

Quadruple-compound engines, two cranks, right angles, $\text{F} = 5{\cdot}5\sqrt{p}$.

,, ,, four cranks, $\text{F} = 5{\cdot}2\sqrt{p}$.

Expansive engines, cranks at 90°, and the rate of expansion 5, $\text{F} = 9{\cdot}5\sqrt{p}$.

Turbines, F = 54.

Single-crank compound engines, pressure 80 lbs., $\text{F} = 10\sqrt{p}$.

The shafts of torpedo-boats, destroyers, and fast craft which are run at full speed only occasionally, and for short periods, may be designed by taking F at about a half of the above values.

The Torsional Stiffness of a Shaft is of more importance to the marine engineer to-day than it has ever been before, inasmuch as it is employed as the measure of the torque of the turbine, and from it the horse-power transmitted is measured (*v.* p. 150).

The torsion meter employed on shipboard to obtain the measure of the power of a turbine is really in essence only an instrument for indicating the angle of twist of a definite portion of one of the shafts that previous to fitting in the ship was tested by torque to ascertain the angular displacements corresponding to the magnitude of each application. The following formula is a means for computing what that angular displacement should be *in degrees*, with a shaft whose external diameter is d and the diameter of bore d_1; the twisting moment or torque is T_1 in inch-pounds, and C is the modulus of stiffness, which for solid shafts of best mild steel is 11,750,000, and for hollow shafts 12,150,000. θ is the angle in degrees; L is the length of the shaft under observation in inches, R the revolutions per minute.

$$(a) \qquad \left.\begin{aligned} \theta^\circ &= \frac{584 \times T_1 \times L}{C \times d^4} \\ &= \frac{T_1 \times L}{20{,}120\, d^4} \end{aligned}\right\} \text{for solid shafts.}$$

$$(b) \qquad \left.\begin{aligned} \theta^\circ &= \frac{584 \times T_1 \times L}{C \times (d^4 - d_1^4)} \\ &= \frac{T_1 \times L}{21.147\,(d^4 - d_1^4)} \end{aligned}\right\} \text{for hollow shafts.}$$

(c) The torque on a shaft $= \dfrac{\text{S.H.P.} \times 33{,}000}{2\pi \text{R}}$ foot-lbs.

$$= \frac{\text{S.H.P.}}{\text{R}} \times 63{,}000 \text{ inch-lbs.}$$

From equation (a) $\quad \text{T}_1 = \dfrac{\theta \times 20{,}120\, d^4}{\text{L}} = \dfrac{\text{S.H.P.}}{\text{R}} \times 63{,}000.$

(b) $\quad \text{T}_1 = \theta \times 21{,}147 \left(\dfrac{d^4 - d_1{}^4}{\text{L}}\right) = \dfrac{\text{S.H.P.}}{\text{R}} \times 63{,}000.$

Then $\quad \text{S.H.P.} = \dfrac{\text{R}}{\text{L}} \times \dfrac{d^4}{3{\cdot}13}$ for solid shafts,

and $\quad \text{S.H.P.} = \dfrac{\text{R}}{\text{L}} \times \dfrac{d^4 - d_1{}^4}{2{\cdot}98}$ for hollow shafts.

Example.—Find the value of θ for a shaft 9 inches diameter and 6 inches bore when transmitting 5,000 S.H.P. at 350 revolutions per minute. The length under observation is 40 inches.

Here the torque $= \dfrac{5{,}000}{350} \times 63{,}000$, or 900,000 inch-lbs.

Equation (b), $\quad \theta = \dfrac{900.000 \times 40}{21{,}147\,(6{,}581 - 1{,}296)} = 0{\cdot}323$ of a degree.

Bending Moment.—If a force is acting on a shaft tending to bend it only, its effort is called the *bending moment*, and is measured by multiplying the force by the distance at which it acts from the support of the shaft.

If the shaft is overhung like a cantilever, and a force P is applied at a distance L from the point of support,

$$\text{The bending moment} = \text{P} \times \text{L} \quad - \quad - \quad - \quad (1)$$

If supported on two bearings, whose distance apart is L, and a force P is applied at a point *midway* between these two bearings,

$$\text{The bending moment} = \frac{\text{P} \times \text{L}}{4}. \quad - \quad - \quad - \quad (2)$$

If the bearings are long—that is, exceeding the diameter of the shaft in length, and are also strong and rigid, so that the shaft is *held* by them sufficiently to prevent flexure taking place in the bearing,

$$\text{The bending moment} = \frac{\text{P} \times \text{L}}{8}. \quad - \quad - \quad - \quad (3)$$

Since, however, few shafts are so secured as to comply with these conditions exactly, any shaft supported in strong bearings not less than 1 diameter long, and whose distance apart does not exceed 10 diameters, and which has to work freely in its bearings, may be treated as partly complying with these conditions, and

$$\text{The bending moment} = \frac{P \times L}{6}. \qquad (4)$$

When the bearings are not rigid enough to prevent flexure of the shaft within them, L must be measured from the centres of the cap bolts, so that where each cap is held down by a pair of bolts L is measured to the centres of bearings. If the caps and bearings are strong and rigid enough to resist any tendency to bend by the action of the shaft, L may be measured from the edge of the bearing or cap. If the bearing is fitted with brasses, which project beyond the cap and bed so much as to receive little or no support from them, L must still be measured from the edge of cap. In a few words, the distance must be measured from what would be the actual points of support if it is bent by severe pressure.

Resistance to Bending.—The strength of a circular section shaft to resist bending is only half of that to resist twisting. If M is the bending moment in inch pounds, and d the diameter of the shaft in inches,

$$M = \frac{\pi d^3}{32} \times f; \text{ and } d = \sqrt[3]{\frac{M}{f} \times 10{\cdot}2}$$

f is a factor which depends on the material of which it is composed, and the value may be based on the fact that it is in tension and compression. The only shafts in a marine engine which are subject to *bending only* are some weigh-shafts having double-ended levers, similar to the side levers of paddle-wheel engines, and their diameter is determined from other considerations than that of mere strength; but with them, as with the crossheads of return connecting-rod engines, care should always be taken that the size suitable for good working in the bearings should be sufficient for strength.

Equivalent Twisting Moment.—When a shaft is subject to both twisting and bending simultaneously, the combined stress on any section of it may be measured by calculating what is called the *equivalent twisting moment*—that is, the two stresses are so combined as to be treated as a twisting stress only of the same magnitude, and the size of shaft calculated accordingly. Professor Rankine gave the following solution of the combined action of the two stresses (*vide* Rankine, *Rules and Tables*, p. 227):—Let T be the twisting moment on a shaft when M is the bending moment on a section, then taking T_1 as the equivalent twisting moment,

$$T_1 = M + \sqrt{M^2 + T^2}.$$

Example.—To find the diameter of a section of a shaft at which the bending moment is 40,000 inch-pounds, when the twisting moment is 250,000 inch-pounds. The shaft of steel $f = 7500$ lbs.

Here

$$T_1 = 40{,}000 + \sqrt{40{,}000^2 + 250{,}000^2}$$
$$= 40{,}000 + 10{,}000\sqrt{4^2 + 25^2}$$
$$= 293{,}170 \text{ inch-pounds.}$$

$$d = \sqrt[3]{\frac{T}{f} \times 5{\cdot}1} = \sqrt[3]{\frac{293{,}170 \times 5{\cdot}1}{7{,}500}} = 5{\cdot}84 \text{ ins.}$$

Crank Shafts.—These shafts are subject always to twisting, bending, and shearing stresses; the latter are so small compared with the former that they are usually neglected directly, but allowed for indirectly by means of the factor f, as already stated.

The two principal stresses vary throughout the revolution, and the maximum equivalent twisting moment can only be obtained accurately by a series of calculations of bending and twisting moments taken at fixed intervals, and from them construct a curve of strains.

Curve of Twisting Moments.—The twisting moment at any position of the crank is equal to the pressure on the piston multiplied by the distance intercepted by a line through the connecting-rod on a line at right angles to centre line through centre of cylinder.

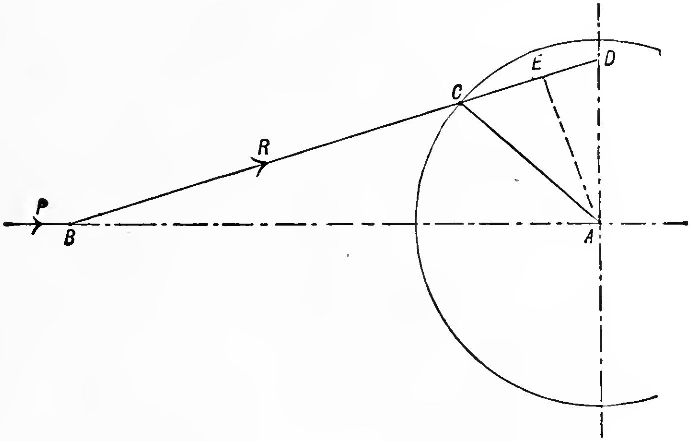


Fig. 102.

Let A B (fig. 102) be the centre line of the engine through the cylinder and shaft centres, A C the position of the crank, B C the connecting-rod, and A D a line at right angles to A B. Produce B C to cut the line A D, and drop from A a line A E perpendicular to B C. P is the load on the piston, and R is the thrust on the connecting-rod. It will easily be proved that the angle D A E is equal to the angle A B D, called for convenience α. Then

$$P = R \cos \alpha; \text{ and } AE = AD \cos \alpha.$$

$$\text{The twisting moment} = R \times AE = R \times AD \cos \alpha = P \times AD.$$

Let the twisting moment be calculated at equal intervals of say 10° of angular movement of the crank, so that in the whole revolution there will be 36 observations, or 18 in the half revolution. Draw a line AB (fig. 103), and divide it into 18 equal parts, $A a_1$, $a_1 a_2$, &c.; erect at these points perpendiculars, and cut off parts $a_1 b_1$, $a_2 b_2$, &c., to represent the value of the twisting moments at each corresponding position of the crank to a suitable scale. Through the points b_1 b_2, &c., draw a curve, which represents the curve of strain on the shaft during the forward movement of the piston; by producing A B, and going through a similar operation for the second half of the revolution, the curve of strain during the backward movement of the piston can be obtained.

Divide the area enclosed between this curve and the line A B by the length of A B, and the quotient is the *mean* twisting moment, and represented by A M in fig. 103, so that the rectangle A M N B is equal in area to the figure A B C.

The value of A M may be calculated by taking a mean of the values of a_1b_1, a_2b_2, etc.

When there are two engines—that is, two pistons operating on one shaft —the combined twisting moment is found by drawing the curve of twisting

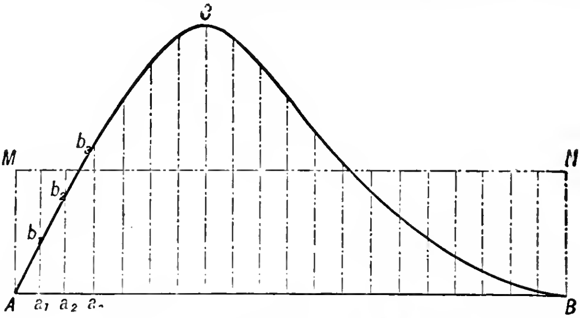


Fig. 103.—Curve of Twisting Moments.

moments of each crank separately, transposing that of one on that of the other in a position corresponding to the relative position of the cranks. In fig. 103*a* A C B is the curve of strain on one crank, and $A_1C_1B_1$ that on the other, which is at an angle with it of degrees represented by AA_1. The combined twisting moment at any period *a* is represented by *a d*, which is equal to *a b* + *a c*, and the dotted curve C*d*C_1, etc., represents the curve of combined twisting moments.

The maximum twisting moment will be at the point where the curve is

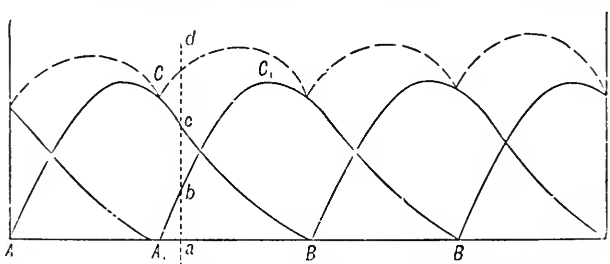


Fig. 103*a*.—Curve of Combined Twisting Moments.

highest, and the ordinate may be measured and its value found by referring to the scale to which the curve is drawn. The mean twisting moment may be found by measuring the area included between the dotted curve and the base line and terminal ordinates, and dividing by the base line, or by taking a mean value from the ordinates as before.

If there are three engines, a similar operation will indicate the maximum twisting moment.

There is another, and perhaps a better, method of showing the curve of torque whereby the magnitude of the twisting moment at any angle throughout the revolution is found by taking the length of the radial line intercepted between the crank-pin circle and the curves, which are constructed by using that circle as a reference instead of a base line. Fig. 104 is a good example of this method, as prepared by Professor Jamieson, for a triple-compound three-crank engine working under ordinary conditions as given below, and will be found instructive.

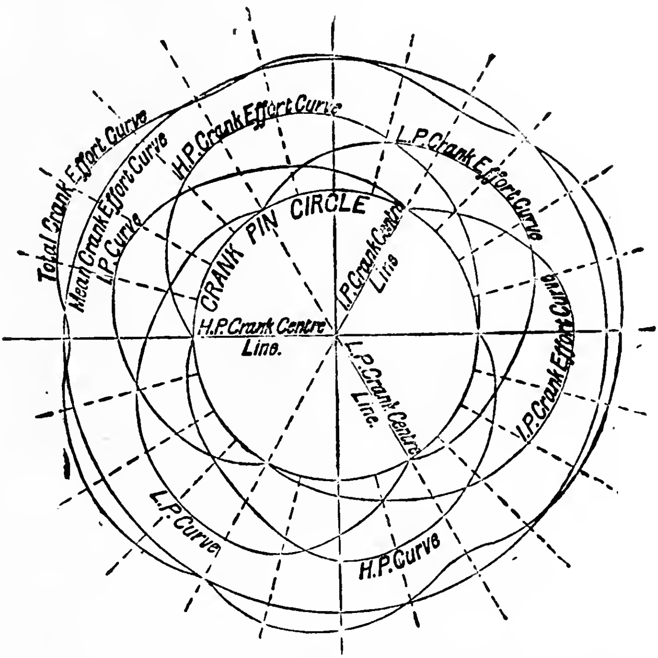


Fig. 104.—Crank Effort Diagram of a Triple-Expansion Engine.

Ratio of expansion, 10·4.

Length of connecting-rod = 9 feet. Stroke = 4·5 feet.

Mean efficiency of steam, or ratio of area of work in cylinder to full theoretical diagram 55 per cent.

	H.P.	I.P.	L.P.
Cylinder's diameter,	28″	46″	77″
Area,	615·8	1661·9	4656·6
Ratio,	1	2·80	7·56
Mean pressures, lbs. per sq. in.,	67·6	28·2	9·7
Range of temperatures, Fah.,	64·3°	74·9°	80·6°

Steam, 164 lbs ; Vacuum, $26\frac{1}{2}$ ins. ; Receivers, 52 and 5 lbs. ;
Revolutions, $62\frac{1}{2}$ per minute.

Cut-off H.P. = $33\frac{1}{2}$ ins.	I.H.P. = 710 L.P.
,, I.P. = ,,	,, = 799 I.P.
,, L.P. = ,,	,, = 764 H.P.
	2,273 total I.H.P.

The pressures at the different points may be taken from actual indicator diagrams, or by constructing steam diagrams from the conditions under which the engine is to work.

The bending moment on a section of the shaft will vary exactly with the pressure on the crank-pin, and to find the maximum equivalent twisting moment on a section, it is only necessary to construct a secondary curve from the formula $T_1 = M + \sqrt{M^2 + T^2}$ between the point of maximum twisting and that at which the pressure on the piston is greatest.

When steam is not cut off in the cylinder before 0·4 of the stroke, the maximum load on the piston may be used to calculate the bending moment. which is to be combined with the maximum twisting moment to find the maximum equivalent twisting moment. Only when steam is cut off earlier than this does the point of maximum equivalent twisting moment differ much from the point of maximum twisting.

Momentum of Moving Parts.—In making these calculations it has been assumed that the moving parts, such as the piston and rods, have no effect on the force exerted on the shaft; but this is never strictly true, for since these parts are of considerable weight, a part of the energy of the steam is absorbed at the commencement of the stroke in overcoming their inertia, and consequently the load on the crank-pin is less then than is represented on the curves. Again, towards the end of the stroke the *momentum*, or energy thus stored in these moving parts, is given out on the crank-pin, and causes larger loads on it than shown by the curve. The further consideration of the effect of the inertia and momentum of the moving parts will be found in Chap. xxix. It is sufficient to say here that the general tendency is to modify the stresses on the crank shaft, as also those of the connecting-rods, piston-rods, etc., so that if any of these are strong enough to withstand the stresses due to external forces, they are sufficiently strong for the engines when moving. Indeed, it is only at very high speeds that momentum need be taken seriously into account.

The following Table (xxxiv.) gives the relation between the maximum and mean twisting moments of engines working under various conditions, the momentum of the moving parts being neglected :—

TABLE XXXIV.

Description of Engine.	Steam cut-off at	$\frac{\text{Max. Twist.}}{\text{Mean Twist.}}$
Single-crank, - - - - - - -	0·2	2·625
,, - - - - - - -	0·4	2·125
,, - - - - - - -	0·6	1·835
,, - - - - - - -	0·8	1·698
Two-cylinder, cranks at 90°, - - - - -	0·1	1·872
,, ,, - - - - -	0·2	1·616
,, ,, - - - - -	0·3	1·415
,, ,, - - - - -	0·4	1·298
,, ,, - - - - -	0·5	1·256
,, ,, - - - - -	0·6	1·270
,, ,, - - - - -	0·7	1·329
,, ,, - - - - -	0·8	1·357
Three-cylinder compound, three cranks at 120°,	H.P. 0·5, L.P. 0·66	1·40
,, triple, ,, ,,	0·6	1·15
Four-cylinder quadruple, four cranks at 90°,	0·6	1·26

Overhung Crank.—The simplest form of crank is that known as the overhung crank, such as is usually fitted in mill engines, but hardly found now in marine engines. The shaft projects beyond the bearing, and has keyed to its end a lever or disc, in which is secured the crank-pin.

The pin is subject to bending and shearing forces, due to the thrust on the connecting-rod. The maximum bending moment on the part of the pin close to the crank is found by multiplying the greatest thrust of connecting-rod by the distance to the centre of the connecting-rod.

If R is the thrust of the connecting-rod, and l the length of the pin, then

$$* \text{ Bending moment on crank-pin } = \frac{R \times l}{2},$$

$$\text{and diameter of pin } = \sqrt[3]{\frac{R \times l}{2} \times \frac{10 \cdot 2}{f}} = \sqrt[3]{\frac{R \times l}{f} \times 5 \cdot 1}.$$

Example.—To find the diameter of the crank-pin whose length is 14 inches

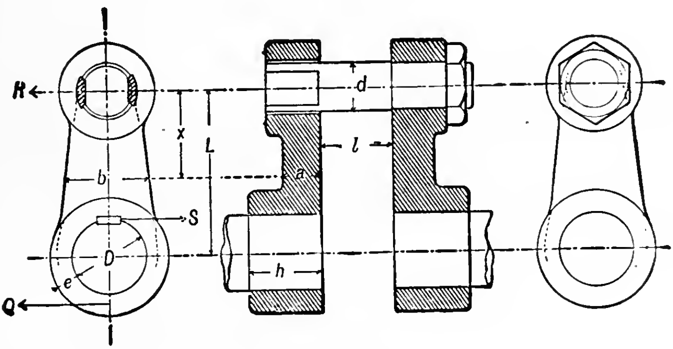


Fig. 105.—Cranks of Paddle-wheel Engine.

and the thrust of connecting-rod is 125,000 lbs., f being of steel and taken at 9,000 lbs.

$$\text{Diameter} = \sqrt[3]{\frac{125{,}000 \times 14}{9{,}000}} \times 5 \cdot 1 = 9 \cdot 97 \text{ inches.}$$

The crank-arm (fig. 105) is to be treated as a lever, so that if a is the thickness in direction parallel to the shaft axis, and b its breadth at a section x inches from the crank-pin centre, then

$$\text{Bending moment M at that section} = R \times x,$$

and
$$\frac{a \times b^2}{6} = \frac{M}{f},$$

or
$$a = \frac{6\,M}{b^2 \times f}.$$

If a crank-arm were constructed so that b varied as $\sqrt{x}$ (as given by the above rule), it would be of such a form as to be inconvenient of manufacture, and consequently it is customary in practice to find the maximum value of

* For other conditions as to size of pins, *vide* p. 309.

b, and draw tangent lines to the curves at the points; these lines are generally, for the same reason, tangential to the boss of the crank-arm at the shaft.

The bending moment decreases as the distance from the crank-pin decreases, while the shearing stress is the same throughout the crank-arm; consequently this latter stress is large compared with the bending stress close to the crank-pin, and so it is not sufficient to provide there only for bending stresses. The section at this point should be such that, in addition to what is given by the calculation from the bending moment, there is an extra square inch for every 8,000 lbs. of thrust, on the connecting-rod. Moreover, the crank-arm is subject to twisting from the action of the pin; strictly speaking, therefore, it should be calculated from the formula $T_1 = M + \sqrt{M^2 + T^2}$.

The length of the boss h, into which the shaft is fitted, is from 0·75 to 1·0 of the diameter of the shaft, and its thickness e must be calculated from the twisting stress $R \times L$.

The crank turns the shaft (fig. 105) by exerting a force S on the key, whose centre of effort is on the circumference, and therefore at a distance of half the diameter from the axis of the shaft, so that

$$S \times \frac{D}{2} = R \times L; \qquad \text{or } S = 2\,R \times \frac{L}{D}.$$

If the crank is loose, the area of the section of the key parallel to the shaft must therefore not be less than $S \div 10{,}000$ lbs. And the load on the section of the crank-boss opposite the key is

$$Q = S - R = R\left(\frac{2\,L}{D} - 1\right).$$

The stress on the section of the boss crossways is T, so that

$$T\frac{D + e}{2} = R \times L; \qquad \text{or } T = 2\,R\,\frac{L}{D + e}.$$

The stress on this section should not exceed 9,000 lbs.

To avoid a complicated expression, it is convenient to assume a relation between h and e, and to substitute the value of e thus found in the above expression. The value of $\frac{h}{e}$ in practice varies from 2, when there is not much space for the crank, to 3, when there is ample room.

Example.—To find the section of the boss of a wrought-steel crank 8 inches long; the pressure on the crank-pin is 54,000 lbs., the diameter of the shaft 10 inches, and $\frac{h}{e}$ assumed at 2·2. Stroke of piston 60 inches.

Here assume

$$e = \frac{8}{2{\cdot}2} = 3{\cdot}67 \text{ inches.}$$

$$T = 2 \times 54{,}000 \frac{30}{10 + 3{\cdot}67} = 237{,}015 \text{ lbs.}$$

$$\text{Area} = 237{,}015 \div 9{,}000 = 26{\cdot}33 \text{ square inches.}$$

And since $$h = 8 \text{ inches}, \; e = \frac{26{\cdot}33}{8} = 3{\cdot}3 \text{ inches.}$$

The cranks of marine engines are always of steel, wrought or cast, and generally of the same materials of which the shaft is made, so that the length and thickness of boss may bear a constant relation to the diameter of the shaft.

When $h = D$, then $e = 0{\cdot}32$ D.
,, $h = 0{\cdot}9$ D, ,, $e = 0{\cdot}34$ D.
,, $h = 0{\cdot}8$ D, ,, $e = 0{\cdot}35$ D.
,, $h = 0{\cdot}7$ D, ,, $e = 0{\cdot}36$ D.

The crank-eye or boss into which the pin is fitted should bear the same relation to the pin that the boss does to the shaft.

Cranks are always shrunk on to both shaft and pin, and when this operation is carefully and well done, a key to the latter is almost unnecessary, and some engineers have latterly omitted to fit one to even very large pins; some engineers simply drill a hole half into the shaft and half into the crank, and drive into it a steel pin so as to answer the purpose of a key.

The diameter of the shaft end on to which the crank is fitted should be $1{\cdot}1 \times$ diameter of the journal. Overhung cranks are never fitted now to screw engines, as they often proved to be very unsatisfactory, from the fact of the whole of the pressure coming on one bearing, and the whole of the bending and twisting stresses being taken by one crank and journal.

Paddle-shafts.—The cranks of a paddle-wheel engine (fig. 105) are still often overhung, and in the case of double engines, the arm to which the pin is secured is the one fitted to the intermediate shaft; the pin fits loosely into an eye on a crank or disc secured to the paddle-shaft, and so drives this latter shaft. The effect of this arrangement is to give a very equable strain to the paddle-shaft, for the pressure of the pin is always at right angles to the crank on the paddle-shaft; and in smooth water the power of each engine is very nearly equally divided between the two wheels, and the *bending* action on the *paddle*-shaft never exceeds half that due to its own cylinder, for when near the dead points the bending moment is at its maximum, and is wholly taken on the crank-arm to which the pin is secured. For these reasons the shaft to which the arm having the crank-pin secured is fitted must be stronger than the outer shafts, especially when the ship is intended to work in rough water, as it is *liable* then to have to transmit the *whole* twisting force of one engine, and *always* takes, during certain periods of the revolution, the whole bending force from that engine. Hence, if T be the maximum twisting moment from one piston of a double paddle-wheel engine, and M the maximum bending moment from that piston, the

Maximum equivalent twisting moment on the intermediate shaft

$$= M + \sqrt{M^2 + T^2},$$

And maximum equivalent twisting moment on the paddle-shaft

$$= \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

Exception may be taken to the latter, since at times when one wheel is out of water the whole of the twisting force of both engines is transmitted through the shaft of the wheel which is deeply immersed; but when the maximum combined effect of twisting is on this one shaft, the bending moment on the crank-journal is probably less than $\frac{M}{2}$, and is that due to the force found by dividing the maximum twisting moment by the length of the crank, which is approximately $\frac{T \times \sqrt{2}}{L}$; the distance at which this force acts is measured from the face of the crank-arm to the edge of the casting, into which the journal brass is fitted.

Example.—To find the diameter of intermediate and paddle-shafts of a double paddle-wheel engine, having cylinders 80 inches diameter and 60 inches stroke, using steam of 45 lbs. pressure absolute, cutting off at 0·6 the stroke; the distance between the bearing beds being 50 inches.

Maximum effective pressure in the cylinder will be about 40 lbs. per square inch. Hence

Load on piston - - $= 5026 \times 40$, or 201,040 lbs.

Maximum twisting moment $= 201{,}040 \times 30 = 6{,}031{,}200$ inch-lbs.

Maximum bending moment $= \frac{201{,}040 \times 50}{4} = 2{,}513{,}000$ inch-lbs.

(1) Maximum equivalent twisting moment on intermediate shaft

$$= 2{,}513{,}000 + \sqrt{2{,}513{,}000^2 + 6{,}031{,}200^2} = 9{,}056{,}000 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{9{,}056{,}000}{8000}} \times 5{\cdot}1 = 17{\cdot}9 \text{ inches.}$$

(2) Maximum equivalent twisting moment on paddle-shaft

$$= \frac{2{,}513{,}000}{2} = \sqrt{\left(\frac{2{,}513{,}000}{2}\right)^2 + 6{,}031{,}200^2} = 7{,}417{,}500 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{7{,}417{,}500}{8000}} + 5{\cdot}1 = 16{\cdot}8 \text{ inches.}$$

The outer part of a paddle-wheel shaft is subject to twisting and bending from the reaction of the water on the floats, and from bending due to the weight of the wheel itself. The pressure on the float can be found by dividing the twisting moment by the distance to the centre of the pressure of the float from the shaft axis in inches. It is practically sufficiently accurate to measure to the centre of fixed floats, and to gudgeons of feathering floats.

For example, the reaction of the water on the floats of the engine in the preceding example, whose radius to float centres is 140 inches, will be found

Pressure on floats $= \frac{6{,}031{,}200}{140}$, or 43,080 lbs.

The twisting moment on the shaft is the same at the outer bearing as at the inner, and is 6,031,200 inch-lbs. The weight of the wheel is 20 tons, or 44,800 lbs., and the distance of its centre from the bearing is 40 inches.

* Maximum bending moment

$$= (44{,}800 + 43{,}080) \times 40 \text{ inches} = 3{,}515{,}200 \text{ inch-lbs.}$$

* In smooth water the bending force is really the *resultant* of the weight and reaction on the floats, and may be taken $= \sqrt{\text{weight}^2 + \text{reaction}^2}$.

Maximum equivalent twisting moment

$$= 3,515,200 + \sqrt{3,515,200^2 + 6,031,200^2} = 10,515,200.$$

Diameter of shaft

$$= \sqrt[3]{\frac{10,515,200}{8000}} \times 5 \cdot 1 = 18 \cdot 88 \text{ inches.}$$

The outer end of a paddle shaft is subject to alternating stresses due to the weight of wheel, and the inner part to intermittent ones (*v.* Table lxxxiii.).

The crank-shafts of paddle engines are now made in the same way as those of screw engines. Sometimes they are made in one piece from a "solid" forging, and sometimes in one piece "built up" (*v.* fig. 19). Those in very large engines have a separate shaft for each cylinder, coupled as in screw engines.

Crank-shaft of Screw Engines.—In case of the forward crank of a double or treble engine, and the crank of a single engine having two arms, there is the action of one engine only on it. On the *forward* journal and crank-arm there is a twisting action sufficient to overcome the friction, and to drive the eccentrics if fixed in this part, and half of the whole bending moment due to the thrust on the crank-pin. On the *aftward* journal, the other half of the bending moment, and the whole of the twisting moment, except the small portion required as above; this portion is at certain periods of the revolutions so small, that in calculations for the journals it may be neglected.

Then equivalent twisting moment on aftward journal

$$= \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

Strain on forward journal $= \dfrac{M}{2}$.

In multiple-crank engines the aftward crank has not only to resist the action of its own piston, but also to transmit the twisting strains of the forward engines. There will be strains from its own piston, which may be calculated in the same way as those on the forward cranks, and to these must be added the twisting strain of the forward engine.

Let T_2 be the maximum twisting moment on the after engine from its own piston, and M_2 the corresponding bending moment, T_1 the twisting moment on the forward engines at the same period.

Then on the *forward* journal of the after crank, the twisting moment is T_1, and the bending strain $\dfrac{M_2}{2}$, so that—

Equivalent twisting moment on forward journal of after crank

$$= \frac{M_2}{2} + \sqrt{\left(\frac{M_2}{2}\right)^2 + T_1^2}.$$

On the *after* journal of the aftward crank, the twisting moment is $T_2 + T_1$, and the bending moment $\dfrac{M_2}{2}$, so that—

Equivalent twisting moment on after journal of aftward crank

$$= \frac{M_2}{2} + \sqrt{\left(\frac{M_2}{2}\right)^2 + (T_2 + T_1)^2}.$$

The bending moment on the after-arm of the aftward crank will be found by calculating the *maximum force on the crank-pin tending to twist the shaft.*

Let T_n be the maximum combined twisting moment, as found by the methods indicated before, L the length of the crank or half-stroke of piston. Then the maximum twisting force at the crank-pin is $T_n \div L$.

The maximum bending moment at any section of the after crank-arm of the aftward crank, whose distance from the centre of the crank-pin is x inches, is $\frac{T_n}{L} \times x$.

The maximum bending moment on a section of the forward arm of the same crank is $\frac{T_1}{L} \times x$.

Example.—To find the sizes of the parts of a crank-shaft of a double expansive engine of 1000 I.H.P., the length of stroke is 40 inches, the cut-off 0·6, and the stroke and the cranks at right angles. Revolutions 60 per minute. Mean twisting moment of one engine $= \frac{500}{60} \times 63{,}000$. Since the cut-off is 0·6, the ratio of maximum to mean twisting moment is 1·835 (Table xxxiv.); therefore

Maximum twisting moment of one engine

$$= 1{\cdot}835 \times \frac{500}{60} \times 63{,}000 = 963{,}375 \text{ inch-lbs.}$$

Mean twisting moment of both engines

$$= \frac{1000}{60} \times 63{,}000.$$

Ratio of maximum to mean twisting moments is 1·27 (Table xxxiv.); therefore

Maximum twisting moment of both engines

$$= 1{\cdot}27 \times \frac{1000}{60} \times 63{,}000 = 1{,}333{,}500 \text{ inch-lbs.}$$

Maximum turning force on forward pin

$$= \frac{963{,}375}{20} = 48{,}168 \text{ lbs.}$$

Maximum turning force on aftward pin

$$= \frac{1{,}333{,}500}{20} = 66{,}675 \text{ lbs}$$

Assuming the distance between the bearings on which the brasses are bedded to be 30 inches,

The maximum bending moment on each of the two forward journals

$$= \frac{48{,}168 \times 30}{8} = 180{,}630 \text{ inch-lbs.}$$

That on the two journals of aftward crank

$$= \frac{66{,}675 \times 30}{8} = 250{,}000 \text{ inch-lbs.}$$

Then diameter of foremost journal

$$= \sqrt[3]{\frac{180{,}630}{8000}} \times 10\cdot2 = 6\cdot13 \text{ inches.}$$

The maximum equivalent twisting moment on after journal of forward crank

$$= 180{,}630 + \sqrt{180{,}630^2 + 963{,}375^2} = 1{,}160{,}630 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt{\frac{1{,}160{,}630}{8000}} \times 5\cdot1 = 9\cdot04 \text{ inches.}$$

The maximum equivalent twisting moment on fore journal of aftward crank

$$= 250{,}000 + \sqrt{250{,}000^2 + 963{,}375^2} = 1{,}245{,}000 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt[3]{\frac{1{,}245{,}000}{8000}} \times 5\cdot1 = 9\cdot25 \text{ inches.}$$

Maximum equivalent twisting moment on aftermost journal

$$= 250{,}000 + \sqrt{250{,}000^2 + 1{,}333{,}500^2} = 1{,}606{,}700 \text{ inch-lbs.}$$

Diameter of journal

$$= \sqrt[3]{\frac{1{,}606{,}700}{8000}} \times 5\cdot1 = 10\cdot1 \text{ inches.}$$

The aftermost crank-arm will be 11 inches across the face; to find its thickness 18 inches from the pin.

Bending moment at that section = 66,675 × 18 = 1,000,000 inch-lbs.

$$\text{Thickness} = \frac{6 \times 1{,}200{,}150}{11^2 \times 8000} = 7\cdot44 \text{ inches.}$$

In actual practice the crank-shaft would not be made with the four journals all of different diameter, but some engineers make the shafts partly in accordance with theory, by arranging the two forward journals of the same diameter, and the two aftward journals of the same diameter; that is, for the example given above, the journals of the forward crank would be each 9·04 inches diameter, and those of the aftward one 10·1 inches diameter.

Example.—To find the dimensions of the crank-shaft of a single engine, whose cylinder is 30 inches diameter and stroke 50 inches, the steam used is 65 lbs. per square inch absolute pressure, and the cut-off at 0·3 the stroke. The distance between foundation facings for shaft brasses is 40 inches. The connecting-rod is 100 inches long. Back pressure and loss at piston are 5 lbs.

The maximum pressure on the piston is 60 × 706 = 42,360 lbs.

The maximum twisting moment occurs just at the cut-off in this case, and is 42,360 × 24, or 1,016,640 inch-lbs.

The bending moment on each journal at that period $\frac{42{,}360 \times 40}{8}$ or 211,800 inch-lbs.

The bending moment on the after arm, at a distance of 22 inches from the centre of crank-pin, is 42,360 × 22, or 931,920 inch-lbs.

Diameter of fore journal

$$= \sqrt[3]{\frac{211,800}{8000}} \times 10{\cdot}2 = 6{\cdot}46 \text{ inches.}$$

Maximum equivalent twisting moment on after journal

$$= 211,800 + \sqrt{211,800^2 + 1,016,640^2} = 1,250,200 \text{ inch-lbs.}$$

Diameter of after journal

$$= \sqrt[3]{\frac{1,250,200}{8000}} \times 5{\cdot}1 = 9{\cdot}27 \text{ inches.}$$

If the crank-arm is 10 inches wide at the face, then thickness of crank-arm at 22 inches from pin $= \frac{6 \times 931,920}{10^2 \times 8000} = 7$ inches.

The bending moment at the centre of the pin of a solid or rigidly built up crank-shaft is $\frac{R \times L}{8}$.

Note.—A solid shaft, or one whose continuity of strength is unbroken from end to end, is treated, so far as bending stresses are concerned, as a girder or beam *secured* at its points of support; or as a continuous girder supported at several points when there are more than two journals. Hence, the bending moment in the *middle* between two journals is $\frac{R \times L}{8}$; *and at the points of support* also $\frac{R \times L}{8}$, since change of flexure takes place at a distance $\frac{L}{4}$ from the supports. Marine crank-shafts whose arms are at least 0·7 of the diameter thick, and whose bearings *thoroughly* support the shaft close to the crank-arms, are really subject to little or no bending action *at the journals.*

At and near the end of the stroke the crank-arms are subject to a bending moment applied very suddenly when the steam enters the cylinder, on opening to lead after only slight *compression*. The force should be taken at twice the load on the piston (2 P), and if L_1 is the length of the pin + the thickness of the crank-arm as close to the shaft, then

$$\text{Bending moment} = \frac{2\,P \times L_1}{4} = \frac{P \times L_1}{2}.$$

Crank-shafts are subject to intermittent loads, and therefore to intermittent stresses (*v.* Table lxxxiii.).

And the bending moment on the section of *each* arm caused by this force is $\frac{P \times L_1}{4}$, and *acts at right angles* to the bending force, due to the force on the crank tending to twist the shaft. Hence,

$$b = 6\,\frac{(P \times L_1)}{4\,(a^2 f)} = \frac{3\,P \times L_1}{2\,a^2 f}.$$

In the previous example L_1 may be supposed to be 23 inches, and P is 42,360 lbs., a is 7 lbs.; then

$$b = \frac{3 \times 42{,}360 \times 23}{2 \times 7^2 \times 8{,}000} = 3{\cdot}6 \text{ inches},$$

so that the forward crank-arm must not be less than this thickness at any part.

Crank-shafts for mercantile screw engines, when above 10 ins. diameter, are generally made in duplicate pieces, so that in case of damage to one only a part of the shaft is condemned, and a spare piece can be easily carried on foreign voyages. And also by this plan there is less labour in replacing the damaged part, than if the whole shaft is moved.

Built Crank-shafts.—Shafts above 10 inches diameter are better built up than in one forging, and they can then be made of steel at much less cost than a solid one; indeed, many engineers now make crank-shafts of all sizes on this principle. The crank-arms are usually of the same thickness at the pin as at the shaft, and equal to 0·7 to 0·8 of the diameter of shaft

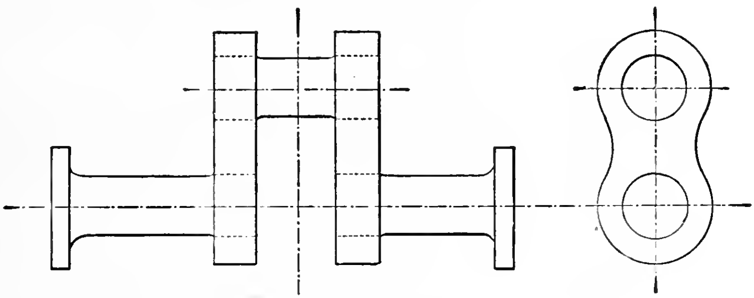


Fig. 106.—Built up Crank-shaft.

journals; the end view, as in fig. 106, shows the usual shape for large cranks—smaller ones are often straight on the sides. Great care is required properly to construct such a shaft so as to be perfectly true when finished, and to have the arms shrunk on sufficiently tight without leaving the metal around the pins, and shaft-ends in such a state of tension as to be dangerous.

The thickness of the metal around the shaft, etc., can be calculated, as before stated for the overhung crank.

The crank-arms are sometimes forged with the shaft-ends, and the pins shrunk into eyes in the arms. This method has advantages, but it is very unsightly, and misses one of the chief merits of a built crank-shaft. Another arrangement is to make the crank-pin and arms in one piece, generally a steel casting, and shrunk it on the shaft-ends or shanks; and sometimes it is secured to the latter by flanges, bolts, etc.

There are a number of other patented forms of crank-shafts, some having the crank-arms of cast steel, and some of forged steel and iron, so arranged as to couple the shafts at the cranks instead of between them.

Fig. 107 shows a piece of crank-shaft as generally made for fast-running engines, and indicate the modifications made when further weight is to be saved, as in naval and other high-speed ships.

Couplings.—It is usual now to have the coupling forged with the shaft instead of keyed on as formerly. The tube shafts of twin-screw engines, however, generally have one coupling keyed on. As a rule, the only stress to which a coupling is subject is due to twisting; hence, if t be the thickness of the flange, and r the distance of any part of it from the centre of the shaft which is subject to a twisting moment T: the section of metal resisting the force is $2\pi r t$; and if f be the stress per square inch on this section, acting at the distance r, then

$$T = 2\pi r t f \times r = 2\pi r^2 . f . t, \text{ that is, thickness of flange} = \frac{T}{2\pi r^2 f}$$

If r is the radius of the shaft subject to twisting only, so that $\frac{\pi r^3}{2} f$ is equal to T. Then

$$\text{Thickness of flange} = \frac{\pi r^3 f}{2} \div 2\pi r^2 f, \text{ or } \frac{r}{4}.$$

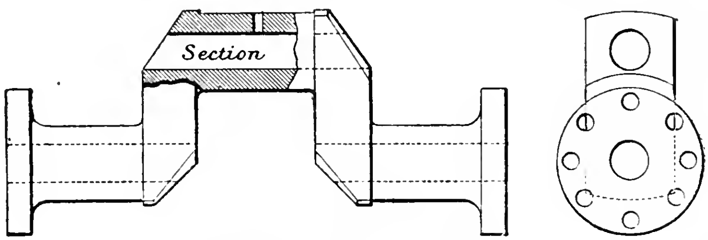


Fig. 107.—Naval Crank-shaft.

From practical considerations the thickness of the flange should not be less than the diameter of the coupling bolts, and since the strength of a coupling is somewhat impaired by the holes drilled for the bolts, it should be about ·27 the diameter of the shaft subject to twisting only.

Coupling Bolts.—When shafts are close coupled, and the bolts are a good fit in the holes, they are subject to a shearing force only, caused by the torque on the shaft; hence, if d be the diameter of the bolts, whose number is n, K is the distance from centre of bolts to centre of shaft, T the twisting moment, and D the diameter of the shaft subject to twisting only. Then

$$T = n\frac{\pi d^2}{4} f \times K; \text{ or } d = \sqrt{\frac{T}{0 \cdot 7854 f . K . n}};$$

but

$$T = \frac{\pi D^3}{16} f.$$

Hence,

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{D}{n \times K}}.$$

If K is always taken at 0·8 × D. Then

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{0·8 \times n}}.$$

Then when there are 5 bolts,

$$\text{Diameter of bolts} = \frac{D}{2}\sqrt{\frac{1}{5 \times 0·8}} = \frac{D}{4}.$$

If 3	bolts, diameter of bolts	=	diameter of shaft	÷	3·10.
,, 4	,, ,,	=	,,		3·58.
,, 5	,, ,,	=	,,		4·00.
,, 6	,, ,,	=	,,		4·38.
,, 7	,, ,,	=	,,		4·73.
,, 8	,, ,,	=	,,		5 06.
,, 9	,, ,,	=	,,		5·37.
,, 10	,, ,,	=	,,		5·67.

The number of bolts in a coupling depends sometimes on circumstances, but usually there should be two, and an allowance of one more for each 2 inches of diameter of shaft, and the above proportions are based on this allowance; but when it is necessary to have the couplings as small as possible the number may be increased, and with the consequent decrease in diameter, the centres of bolts may be nearer to the centre of shaft.

The couplings of a two-crank engine, whose shaft is in duplicate halves at right angles, should have four or a multiple of four bolts; and for those of a three-crank engine, whose shaft is in three duplicate pieces, the number of bolts must be a multiple of three.

With shafts of mild steel, the bolts should be of a harder kind; indeed, as they are subject to shearing stresses only, they may be made with advantage of a steel 40 to 45 tons tensile strength.

Surface of Crank-pins and Shaft-journals.—Measuring, as in the case of gudgeons and crossheads, the effective bearing surface as the diameter multiplied by the length of the bearing, the bearing surface of crank-pins was such that the pressure per sq. in. did not exceed 500 lbs.; this, however, is hardly sufficient for the pins of high-speed engines, which should be such that the pressure per square inch does not exceed $\frac{6,500}{\sqrt{R + 100}}$ lbs.—that is, if d is the diameter of the crank-pin, and l its length, L the maximum recurrent load on piston, R the revolutions per minute, then

$$d \times l = L \div \frac{6,500}{\sqrt{R + 100}}; \text{ or } \frac{L\sqrt{R + 100}}{6,500}.$$

When the brass is recessed, so that it bears only in parts on the shaft, the actual bearing surface should not be exposed to more than 600 lbs. pressure per square inch.

The pins of paddle-wheel engines, owing to the comparatively slow speed of shaft, may be designed, if necessary, to take a pressure of 800 to 900 lbs. per square inch.

The main bearings in which the crank-shaft runs should be such that the pressure never exceeds 600 lbs. per sq. in. in paddle engines, and in screw engines it should not exceed 400 lbs. The main bearings of screw engines, when room admits, should be such that the nominal pressure in lbs. per sq. in. does not exceed $\frac{4,300}{\sqrt{R + 100}}$, measuring the whole of the bearing. If, then, d be the diameter of the journals and l the length of each—

$$d \times l = \frac{L}{2} \times \frac{\sqrt{R + 100}}{4,300}.$$

The length of the crank-pin is from 1 to 1·5 of the diameter, and that of each journal from 1·2 to 1·5 the diameter of the journal. Vertical engines have usually sufficient space for a crank-pin 1·25 the diameter, and each journal 1·5 the diameter. The foremost journal of a compound engine is often made much shorter than the others, to allow the eccentric sheaves to be nearer the centre, so as to come in line with the valve-rod.

Owing to the comparatively small pressure on the crank-pins and journals of three-crank compound, and triple- and quadruple-compound engines, the intermediate journals may be generally somewhat shorter; but the first and last journals should not be materially less than was usual in a two-crank engine.

Drivers.—In order to avoid any of the thrust of the propeller coming on the crank-shaft and its bearings, the coupling-bolts connecting it to the thrust-shaft are sometimes made without heads, so that they are free to move in and out of the holes in the coupling flange of one of these shafts while held firmly in that of the other. When this is so they should be of larger diameter than the ordinary coupling-bolts, and the part projecting from the face of the flange into which they are secured should be larger still, so as to form a shoulder, against which they may be tightened up, and give the necessary strength to resist bending. The faces of the flanges when thus loosely coupled should be from $\frac{1}{4}$ to $\frac{1}{2}$ inch apart. It was found necessary, generally, to provide means for lubricating these drivers, especially in heavily armoured and high-powered warships with single screws. Drivers are still necessary in all ships which are liable to change shape from varied loading, as in cargo steamers in heavy seas, and in long steamers of light build. As a rule, however, it is better to connect the crank and thrust-shafts in the ordinary way with bolts, etc., and to leave such clearance in the bearings and brasses that the crank-shaft may have some longitudinal "play."

Taper Bolts are often used in lieu of the ordinary parallel ones, especially for the shaft next the propeller-shaft, to facilitate their withdrawal. Taper bolts can be used with advantage when the flange is by necessity small, for the screwed end is much smaller than the part at the junction of the two shafts subject to shearing. These bolts also are necessarily a tighter fit in the hole, since the tightening of the nuts draws them farther into it.

Cross-keys are sometimes fitted to couplings. Half the key is bedded into a recess in the face of each flange, and so it takes the shearing stress from the bolts.

Since with a number of bolts or drivers it is possible, from wear or bad workmanship, that the load is taken only by a part of them, it is usual to provide an excess of strength. This provision can be conveniently effected by proportioning them to the diameter of the *crank-shaft* as if it were subject to twisting only.

Propeller-shafts,* sometimes called "screw" shafts, and sometimes "tail-end" shafts. The propeller shaft is subject always to the torque of the engine, and to bending due to the weight of the propeller. In rough weather, when the ship is pitching, the strains are increased and often become very severe; for when the screw is partially immersed, in addition to the twisting moment by the reaction of the water acting on one side only, there is a bending stress as on a paddle-shaft; besides which the momentum of the screw when pitching also adds severe bending stresses, all of which are moreover alternating.

In still water the bending moment on the shaft is the weight of the screw multiplied by the distance of its centre from the stern bush. To provide for the strains in rough weather, the bending moment should be taken at twice this value; for ships which may cross the Atlantic at any time in ballast trim, even this is insufficient to produce a shaft sufficiently large to last a satisfactory length of time, inasmuch as the material of a shaft working under such conditions is subject to alternating stresses of considerable intensity, which tend to degrade it and render it unfit to resist such stresses as come on it. The diameter of the screw is also an important factor in determining the size of screw shafts of ships subject to rough weather.

Hence, if T is the maximum twisting moment on the crank-shaft, W the weight of the propeller in pounds, and L the distance of its centre from the stern bush,

$$\text{Maximum bending moment} = 2\,W \times L;$$

and

$$\text{Maximum equivalent twisting moment } T_1 = 2\,W \times L + \sqrt{(2\,W \times L)^2 + T^2};$$

and as before,

$$\text{Diameter of screw-shaft} = \sqrt[3]{\frac{T_1}{f} \times 5{\cdot}1}.$$

Example.—To find the diameter of the screw-shaft for an engine whose maximum twisting moment is 1,333,500 inch-lbs. The weight of the screw is 6000 lbs., and the distance of its centre from stern bush is 20 inches.

The max. bending moment

$$= 2 \times 6000 \times 20 = 240{,}000 \text{ inch-lbs.}$$

The max. equivalent twisting moment

$$= 240{,}000 + \sqrt{240{,}000^2 + 1{,}333{,}500^2} = 1{,}594{,}000 \text{ inch-lbs.}$$

Diameter of shaft

$$= \sqrt[3]{\frac{1{,}594{,}000}{6000} \times 5{\cdot}1} = 11{\cdot}2 \text{ inches.}$$

It is such a very serious matter when the screw-shaft breaks, that it should always be of ample size, and for ships in the Atlantic trade it should be specially strong. It was usual to make it the same diameter as the crank-shaft, but in some ships even this is not sufficient, and it is now not at all an unusual thing to make them 20 per cent. stronger than the crank-shaft.

Where the screw is fitted in a "banjo" frame for lifting above water when the ship is under sail, the shaft is, of course, nearly wholly free from bending stresses.

* The outboard shafts of naval ships and large express steamers are generally hollow, and for stiffness are of large diameter, with the hole as much as $0{\cdot}7 \times$ the external diameter.

Outer Bearing.—It was formerly customary to provide an outer bearing on or in the rudder-post, for the extreme end of the screw-shaft to rest upon; but since the rudder-post practically gives no support sideways, and a very precarious one in any direction, the practice is now obsolete. Also, it was found that when ships so fitted touched the ground with the heel, the screw-shaft was often bent, and sometimes dangerously so. The strongest argument in favour of this outer bearing is, that it prevents the loss of the screw when the shaft is broken; but if the shaft is broken, and the ship has to depend on the sails, it is better perhaps to be without the screw: and if the shaft is broken diagonally, as is often the case, and the screw is caused to revolve from the motion of the ship, there is great risk of splitting the stern tube. If there is no outer bearing, and the fracture is well within the tube, the screw will not be lost, but will go back until the shaft-end butts against the rudder-post, and revolves then without danger. If the shaft breaks close to the propeller, and there is an outer bearing, the danger of damage to the rudder-post and rudder is very great indeed, from the wrenching of the bearing on the propeller falling out.

Screw-shaft End.—The shaft end fitted into the screw boss should be turned to a taper of $\frac{3}{4}$ inch to the foot; if the taper is less than this, as was sometimes the case, extreme difficulty is experienced in getting the screw off. The screw should be secured by a key extending the whole length of the boss, and driven into place after the screw is thoroughly well driven on. The screw is retained in place by a nut, the screw-thread of which is the reverse hand of that of the screw itself. A tail key through the shaft end was preferred by some engineers as a means of retaining the screw in place; but although it is a very safe plan, it is not so convenient as the nut. When a nut is employed a safety key or pin is fitted in rear of it (*v.* fig. 173), or else a set-screw or other simple means of locking it is used. These nuts are now often of cast steel or bronze, and made so as to cover the end of the shaft. When the propeller is of bronze a "cap" nut of the same material is necessary. Such nuts are secured by set-screws in the end "out of centre," or set-screws in the boss-fitting in recesses in the side of the nut (*v.* fig. 174).

Screw-shafts are encased with bronze from the propeller to the inner end of the stern tube in H.M. Navy; in merchant ships this is now frequently the case, although objected to partly on account of the expense, and partly because it prevents examination of the shaft and the detection of flaws, which may extend unobserved until rupture takes place. Bronze casings, on the bearing parts, are used not so much to protect the shaft from corrosion, as to provide a better wearing surface when running on lignum vitæ, and to admit of wear without weakening the shaft thereby. Lloyd's Register now require all casings to be the full length of the stern tube.

When working in sandy water lignum vitæ wears very quickly and grinds away the bronze casing. Ships which are often exposed to this are better without the bronze casing and the lignum vitæ, and should be fitted in lieu with Fenton's white metal, and the shaft either without casing or cased with a hard steel liner, which may be renewed when worn.

The Stern Bush should be of such length that the pressure per square inch (measured as stated for bearings) does not exceed $\frac{500}{\sqrt{R + 100}}$ lbs. This bush has to sustain the weight of the propeller and that of a considerable portion of its shaft; in a seaway it has also added to its load that due to inertia. The above allowance, however, for smooth-water conditions provides a large enough bush for sea-going ones. If d is the diameter of the shaft over the bronze casing, $d_1 = 0{\cdot}9\,d$ that of the shaft under it; l the length of the bearing part of the bush, all in inches, and W the weight of the propeller in pounds, and the length of shaft, whose weight is borne by the bush, as 15 diameters, or $15\,d_1 = 13{\cdot}5\,d$.

$$\text{Weight of shaft} = \frac{\pi\,d_1^2}{4} \times 15\,d_1 \times 0{\cdot}28 = 3{\cdot}3\,d_1^3, \text{ or } 2{\cdot}4\,d^3.$$

Then total weight on bush $= \mathrm{W} + 2{\cdot}4\,d^3$ pounds.

$$\text{Then } l \times d \times \frac{500}{\sqrt{\mathrm{R}+100}} = \mathrm{W} + 2{\cdot}4\,d^3,$$

and
$$l = \left(\frac{\mathrm{W}}{d} + 2{\cdot}4\,d^2\right) \times \frac{\sqrt{\mathrm{R}+100}}{500}.$$

In a general way this total weight on the bush $= 1{\cdot}3$ W for solid shafts, and $1{\cdot}225$ W for hollow ones.

$$\text{Then, length of bush} = \frac{\mathrm{W}\sqrt{\mathrm{R}+100}}{\mathrm{F} \times d},$$

where for solid shafts F = 385, and for hollow shafts F = 408.

Example (1).—What should be the length of the bearing strips of a stern bush for a ship whose screw weighs 25,000 lbs., revolves at 100 per minute, and the shaft is 17 inches over the liner.

$$\text{Length} = \left(\frac{25{,}000}{17} + 2{\cdot}4 \times 289\right)\frac{\sqrt{200}}{500} = 60{\cdot}3 \text{ inches.}$$

Example (2).—For a turbine-driven ship the screw is 6,600 lbs. in weight, the shaft 12 inches over the liner, revolutions are 500.

$$\text{Length} = \frac{6{,}600 \times \sqrt{600}}{409 \times 12} = 33 \text{ inches.}$$

*The stern bush in practice is of a length equal to three to four times the diameter of bore. The stern-shaft should be supported on a bearing in the tunnel when possible; when this is either not possible or inconvenient, it should rest on a bush in the stern tube just abaft the stuffing-box.

In the mercantile marine the screw-shaft, when partly cased with bronze, and running on bushes fitted with lignum vitæ, the brass casings should extend from the screw boss to an inch or two beyond the inner end of bush, and should do so also where the shaft passes through the stuffing-box, and inner bush when there is one. Of late years there has been a tendency to follow the Admiralty method of casing in the whole length covered by the stern tube. In order to make the process as easy and inexpensive as possible, the stern tube is made very short, so that the casing is very little longer than the sum of the lengths of the two liners as formerly fitted.

The Thickness of Brass Casing at bearing parts is $0{\cdot}3$ inch $+ 0{\cdot}035 \times$ diameter. In order to easily withdraw the shaft, the diameter of the inner casing should be $\frac{1}{8}$ inch larger than the outer.

* Lloyd's Register requires it to be at least 4 diameters in length.

Stern Tube.—In the Navy the stern tube is always of bronze, and is within another steel tube secured to the framing of the ship. The lignum vitæ strips are fitted into the tube, either in separate grooves for each strip, or the strips fit in side by side with a brass strip at the top secured by screws to the tube which keys them so as to form a bush. A similar but shorter set of strips is fitted next the stuffing-box. The brass stern tube fits into the stern-post or spectacle piece accurately and tightly, and is secured to the bulkhead by a flange, etc.*

In the mercantile marine the stern tube is nearly always of cast iron, whose thickness is 0·5 inch + 0·08 the diameter. The stern bush of bronze fits accurately into the tube, and the stuffing-box neck ring and gland are lined with brass. There are various ways of fitting the stern tube in place.

The common plan adopted by most engineers is to turn the outer end so as to fit accurately and tightly into the hole bored in the stern forgings or castings, and secure it by a nut screwed on its end. The inner end has a flange, and next it a projecting rim, which is turned to fit in the hole bored in the bulkhead; the flange is bolted to the bulkhead after a liner is fitted between them (*v.* fig. 108).

The tube was sometimes bolted to the stern-post by two lugs cast with it, one above and one below it.

Another plan of fixing the stern tube is to fit its outer end into a recess bored in the stern-post, and secure it by bolts to the bulkhead as before described, and by two strong draw-bolts passing through the flange to a partial bulkhead two or three frame spaces nearer the stern. In this case the stern bush is partly in the tube and partly in the stern-post.

Stern Bushes.—When made of white metal they should be of a thickness = 0·5 inch + 0·03 × diameter. Those fitted with lignum vitæ are of bronze, and formed with a flange, which is secured to the stern tube by screws, which prevent it from turning or coming out. Stern bushes are often made of strong cast iron, with white metal strips driven in, or with it run in as in a main bearing. Such bushes have flanges like the bronze ones. The lignum vitæ is sometimes fitted in strips, as in stern tubes, and sometimes into square holes, the bush being cast as a skeleton to hold the wooden blocks. This latter plan is very convenient for small ships, but not a good one for large ones, as the wood by the continued concussion gets impressed on the cast-iron tube. Lignum vitæ wears best in end grain, especially when it is of inferior quality; when cut from a good tree of large size it wears equally well either way.

Lignum vitæ and white metal strips should be from $\frac{3}{8}$-inch to $\frac{3}{4}$-inch thick, and about three to four times their thicknesses in breadth; they must be bevelled so as to leave free watercourses between them. The brass behind the strips should be 0·04 × diameter in thickness, and the metal ridges between the wooden strips of the same thickness. Sometimes the bronze bush is dispensed with, and the strips fitted into the cast-iron tube as in the brass tube, but this is not good practice.

A pipe is fitted leading from the top of the stern tube to the bulkhead, through which the water may run from the tube so as to cause a fresh supply to enter from the sea, and thereby prevent heating (*v.* fig. 108).

* It is usual, however, to fit a separate bush containing the lignum vitæ strips in all but small ships for convenience of repair.

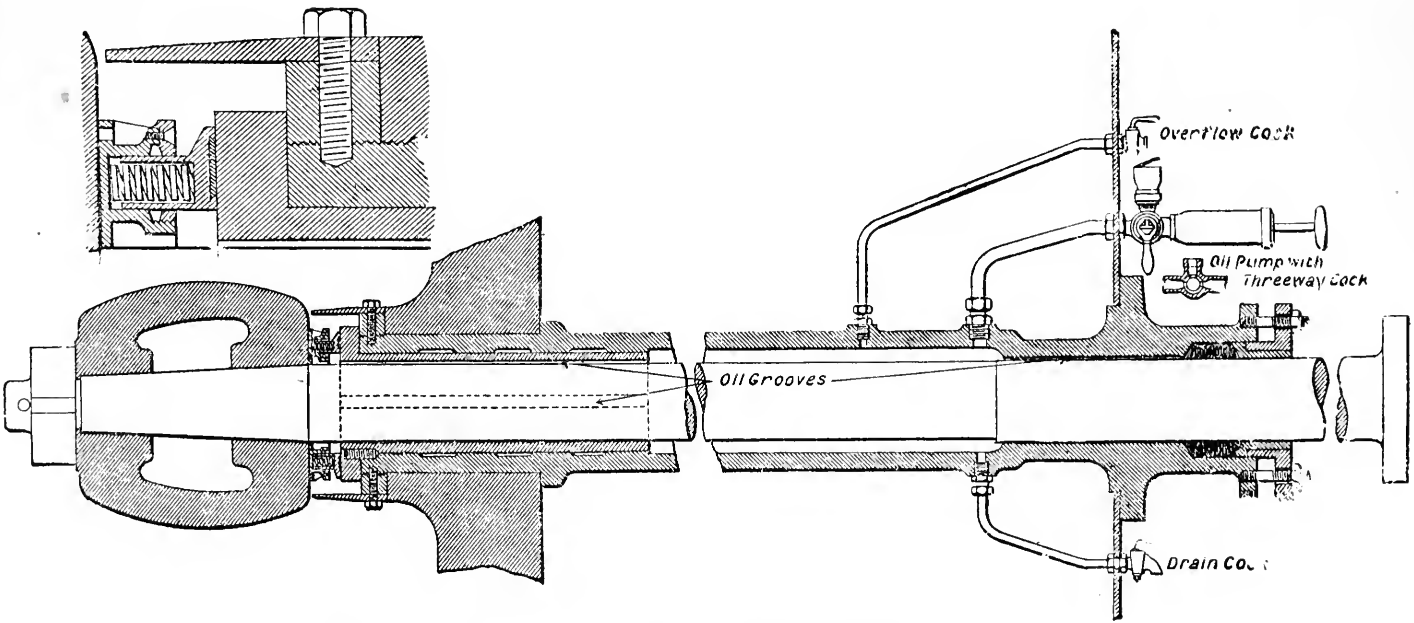


Fig. 108.—Screw-shaft and Stern Tube, Cedervall's Patent for Oil Lubrication.

From time to time attempts have been made to exclude sea water from the stern tube, and lubricate the shaft bearings with oil or soapy water. Ordinary stuffing-box, gland, etc., have been fitted to the tube end, and the lubricant fed through a pipe from the upper deck, so as to give a "head" superior to that of the sea water, and so prevent any leakage inward. No great degree of success has followed this plan, as no provision is made for the wear of the bush. But if the bush did not wear the violent action of the screw in a seaway would inevitably cause leakage. This difficulty has been got over by Mr. Cedervall, who prevents water from entering by the means shown in fig. 108, which have been generally successful in practice.

Thrust-shaft.—Although the crank-shaft was sometimes made with a collar or collars on it, to take the thrust, it is not good practice, especially for large engines. The crank-shaft should be required to take only such loads and motions as are due to the direct action of the pistons, and be free to move around in its bearings without end pressure; and since any longitudinal displacement of the crank-shaft tends to throw abnormal strains on the working parts, it is better to remove all causes of such a derangement. To this end the thrust collars should be on one of the intermediate shafts, and for convenience on that one next the crank-shaft. If possible the thrust bearing should be in the engine-room, and it is for this purpose chiefly that the collars were sometimes on the crank-shaft.

Thrust.—To find the thrust along the shaft of a screw engine, it is necessary to know the speed of the ship and the effective horse-power. The effective horse-power is in this case the power actually employed in producing thrust, and, of course, its relation to the *indicated* horse-power depends on the combined efficiency of the engines and propeller. This ratio may be taken as 0·77 with the best high-speed engines, and 0·68 with the ordinary merchant steamers' machinery. With turbines it is probably 0·8. If P be the pressure in pounds exerted by the propeller against the thrust bearing, and S the speed of the ship in feet per minute, then

$$\text{Work done in moving the ship} = P \times S,$$

and therefore

$$\text{Effective H.P.} = P \times S \div 33{,}000,$$

or

$$\text{I.H.P.} \times f = \frac{P \times S}{33{,}000},$$

then

$$P = \text{I.H.P.} \times \frac{33{,}000}{S} \times f.$$

Now, if K be the speed in knots,

$$S = K \times \frac{6080}{60}.$$

Then

$$P = \text{I.H.P.} \times \frac{326 \times f}{K}.$$

P is called the *mean normal thrust*.

Thrust may be ascertained fairly accurately from the formula—

$$\text{Thrust in pounds } P = \frac{D^2 \times \sqrt{A} \times V^2}{P_r} \times G.$$

D is the diameter in feet, A the acting surface in square feet, V is = pitch × revolutions per second, G = 0·4 to 0·5, and P_r is the pitch ratio.

Example.—To find the thrust on the shafting of an engine whose I.H.P. is 2,000, and the speed of the ship 12 knots, the efficiency 0·66

$$P = 2{,}000 \times \frac{326 \times 0{\cdot}66}{12} = 36{,}166 \text{ lbs.}$$

Now, it will be seen that P varies with the I.H.P., and inversely as the speed, so that the thrust of a particular screw may vary very considerably; for if from some cause the speed is decreased, without a corresponding decrease in the power, the thrust must of necessity increase. This actually occurs in practice, and must be provided for always. The times when the actual thrust exceeds the normal thrust are when the engine first moves, and its power is employed in overcoming the inertia of the ship, also when the ship is towing, and when driving against a head wind or sea. It is also to be noted that the speed of ship means speed *through the water*; for it is on this account that so little strain comes on the moorings of a ship whose engines are working at full speed when in a dock or confined piece of water. In this case it is only at first starting that any great tension is thrown on the moorings, for as soon as the water is set into motion so as to flow past the ship in a steady stream, the power is absorbed in facing the stream and really propelling the ship through the water.

Another cause of variation in the thrust is the variation in the twisting moment, which is, as before shown, very great in certain classes of engines.

The surface exposed to thrust may, however, be calculated from the mean normal thrust, and allowance made for all emergencies. This surface should be such that the pressure per square inch from the mean normal thrust does not exceed 70 lbs.; and for tugboats or ships especially exposed to severe weather, or service analogous to either of these, it should not exceed 50 lbs. Ordinary merchant ships have usually such surface that the nominal thrust does not exceed 60 lbs., and naval ships 45 lbs., as they may have to tow or do analogous work at full power.

Friction Loss at the Thrust Block is not so great as generally supposed by those who devise and patent special means for its reduction. As a matter of fact, with carefully turned steel shafts with multiple collars running against good white metal well lubricated the loss is never more than 1·5 per cent. of the I.H.P. transmitted by the shaft, and generally is as follows:—

The ordinary merchant steamer, . . .	0·4 per cent.
Express steamers and large naval ships, . .	0·5 ,,
High-revolution steamers and naval ships. .	0·65 ,,
Turbine steamers of large size, . . .	0·70 ,,
,, high revolution, . .	1·0 to 1·3 per cent.

The following rule holds good for all:—

$$\text{Loss per cent.} = 0{\cdot}05 \sqrt{\text{revs. per min.}}$$

***Pressure per Sq. In. on a Thrust Bearing** should not exceed $2{,}700 \div \sqrt{Rd + 100}$, R being the revs. per min. and d the diameter of thrust-shaft in inches.

Diameter of Thrust Collars.—Let P be the mean normal thrust, d the

* With Michel bearings the pressure may be as much as $17{,}500 \div \sqrt{R\,d + 100}$.

diameter of the shaft, and D the diameter of the thrust collars, whose number is n, and the pressure 60 lbs. Then

$$P = 60\left(\frac{\pi D^2}{4} - \frac{\pi d^2}{4}\right) n = 47\,(D^2 - d^2)\,n;$$

and

$$D = \sqrt{d^2 + \frac{P}{47\,n}}.$$

The thickness of each collar for mere strength

$$= \frac{P}{n} \div (\pi d \times 1000) = \frac{P}{3142\,d\,n}$$

In practice the thickness of each collar $= 0{\cdot}4\,(D - d)$.

(1) Space between the collars, if rings are of solid brass $= 0{\cdot}4\,(D - d)$.

(2) Space between the collars, if rings are of cast iron faced with bronze or white metal $= 0{\cdot}75\,(D - d)$.

(3) Space between the collars, if rings are of hollow bronze for water to circulate through $= D - d$.

The number of collars depends very much on the size of the engine and the prejudice of the designer. If there are many collars, they are of necessity somewhat small, and although the *chances* are in favour of the majority of them acting efficiently, provision must be made for the contingency of the whole thrust coming on only one of them, and the larger the number of collars, the less able is each one separately to resist the whole thrust. The chief objection to a few collars is, that they are of necessity of comparatively large diameter, and have, therefore, a greater resistance to turning owing to the longer radius, besides a higher speed of rubbing surface; there is also the consideration of cost of forging against large collars. On the other hand, when there are a few large collars a better design of thrust-block is possible, and the rings can be made adjustable without removal.

The number of collars should vary with the size of the shaft, and a very good rule is, that there should be one collar for shafts up to 5 inches diameter, and then an additional collar for every 1·8 inches of diameter beyond this when the collars are large, and the engines slow running; fast-running naval engines require more than this to get sufficient surface, say an additional collar for each additional inch and a quarter. That is,

$$\text{Number of collars for naval and express} = 1 + \frac{d-5}{1{\cdot}25}.$$

$$\text{,,} \qquad \text{,,} \qquad \text{mercantile engines} = 1 + \frac{d-5}{1{\cdot}8}.$$

Michel's Thrust-block, shown in fig. 109, is based on the principle followed by this engineer in all rubbing surfaces, whereby perfect lubrication is obtained and freedom from metallic contact of surfaces ensured. This arrangement requires one shaft collar only; on each side in the block is a horse-shoe-shaped inverted collar, each containing six segments of suitable metal—bronze or white metal—loosely held and pivoted out of centre so as to tip slightly but sufficiently to admit the passage of oil freely as the shaft revolves and maintain oil films between the surfaces, which are sheared continuously. The inventor claims that this resistance to shear is constant and about ½ lb. per square inch, whatever the thrust pressure may be; the heat generated is, therefore,

$$\text{B.T.U.} = \text{number of square inches} \times \text{speed} \div 2 \times 778.$$

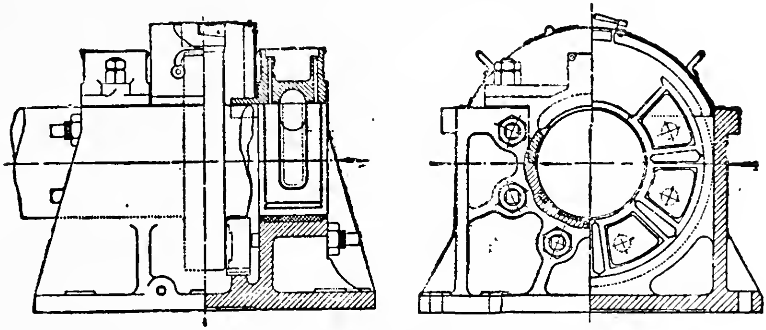


Fig. 109.—Self-contained Michel Marine Thrust Bearing.

In practice the coefficient of friction is only 0·0015, and a pressure of 500 lbs per square inch is not excessive under ordinary working conditions. With highly viscous oil as much as 5 tons per square inch has been maintained satisfactorily.

Fig. 110 is the form of block generally used. Here the horse-shoes fit

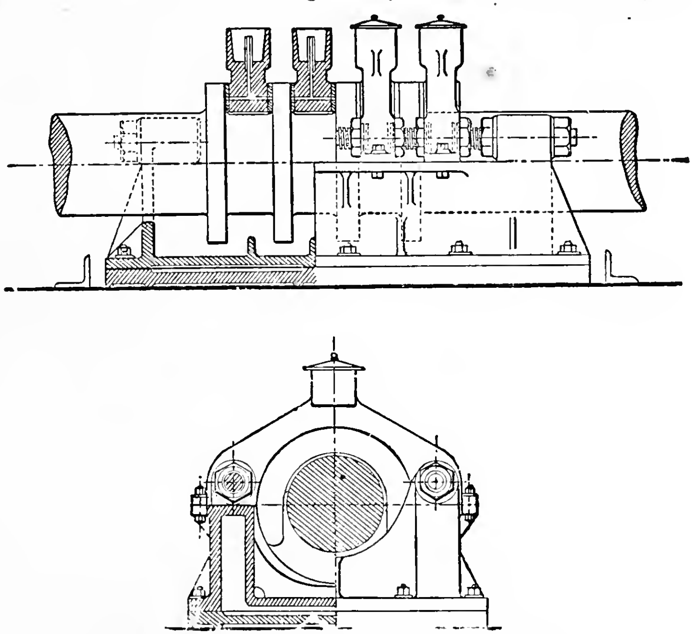


Fig. 110.—Thrust-block with Adjustable Collars.

over two screwed bars, one on either side of the block; nuts are fitted to these bars, so that each collar may be adjusted by its own nuts, or the whole of them by the nuts at the end. A simpler and cheaper method is to turn these side bars so that there is a collar between each pair of shoes instead of a pair of nuts—that is, the bars have a set of collars corresponding to those of the shaft—and take the thrust of the shoes as before.

Both these plans are most successful in practice, in great measure due to the fact that the collars are open and exposed at the top, so as to be easily lubricated and cooled by the air, and to their running in oil, or in a mixture of oil and soapy water contained in the trough below them.

It is most important that a bearing be placed close to the thrust, so that the shaft cannot vibrate and cause uneven pressure over the surface of the collars. The function of the thrust bearing is to take only end pressure. This is particularly the case when designed with horse-shoe rings.

The Length of the Bearings of Tunnel Shafting will depend on the size of the shaft and their distance apart. If d is the diameter and L the distance in inches, the weight of the shaft which each bearing has to support is

$$W = \frac{\pi d^2}{4} \times L \times 0{\cdot}282, \text{ or } 0{\cdot}22\, d^2 \times L.$$

The bearing surface is taken at $l \times d$ square inches, and the pressure should be about 60 lbs. per square inch in smooth water when the rate of revolution does not exceed 100 per minute; taking

$$\text{Pressure per square inch} = 800 \div \sqrt{\text{revs.} + 100}.$$

Then
$$0{\cdot}22\, d^2 \times L = d \times l \times \frac{800}{\sqrt{R + 100}},$$

and
$$\text{Length of bearing} = \frac{d \times L \times \sqrt{R + 100}}{3.640} \text{ inches.}$$

Example.—What should be the length of tunnel shaft bearings in a turbine ship having shafts 6 inches diameter running in bearings 10 feet apart at 800 revolutions per minute?

$$\text{Length of bearing} = \frac{6 \times 120 \times \sqrt{800 + 100}}{3{,}640} = 5{\cdot}93 \text{ inches.}$$

Diameter of Shafts, Rules for, in Practice.—As almost every ship is classed with one or other of the shipping registers, the shafting must in no case be smaller than provided for in the rules of the corporation with which she is classed, and from whom a machinery certificate is necessary. All British ships carrying more than 12 passengers must have a certificate from the Board of Trade, whose surveyor has to report that, *inter alia*, the shafting is fit and proper, etc.; in his case it means that the diameters are not less than given by the Board's Rules.

The Admiralty now generally specify the sizes of the various shafts, which are almost invariably hollow steel.

Board of Trade Rules for Shafts.

For compound condensing engines with two or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}$$

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right)$$

Where S = diameter of shaft in inches.

d^2 = square of diameter of high-pressure cylinder in inches or sum of squares of diameters when there are two or more high-pressure cylinders.

D^2 = square of diameter of low-pressure cylinder in inches or sum of squares of diameters when there are two or more low-pressure cylinders.

P = absolute pressure in lbs. per square inch—that is, boiler pressure plus 15 lbs.

C = length of crank in inches.

f = constant from following table.

Note—Intermediate pressure cylinders do not appear in the formulæ.

For ordinary condensing engines with one, two, or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{3 \times f}}, \qquad P = \frac{3 \times f \times S^3}{C \times D^2}.$$

Where D^2 = square of diameter of cylinder in inches; or sum of squares of diameters where there are two or more cylinders. Other symbols as above

TABLE XXXV.—Board of Trade Factors for Shafts.

For Two Cranks—Angle between Cranks.	For Crank and Thrust Shafts. f		For Tunnel Shaft. f	For Propeller Shaft.* f
90°	For paddle engines of ordinary type multiply constant in this column suitable for angle of cranks by 1·4.	1047	1221	890
100°		966	1128	821
110°		904	1055	768
120°		855	997	727
130°		817	953	694
140°		788	919	670
150°		766	894	651
160°		751	877	638
170°		743	867	631
180°		740	864	629
For Three Cranks. 120°		1110	1295	943

Note.—When there is only one crank the constants applicable are those in the Table opposite 180°.

* The portion of the propeller shaft which is forward of the stern gland, and all the thrust shaft, with the exception of the part enclosed in the thrust bearing, may be the same diameter as the intermediate tunnel shafting.

Lloyd's Rules with regard to Shafting.

The diameters of crank and straight shafts are to be not less than those given by the following formulæ :—

TABLE XXXVI.—Diameters of Shafts—Lloyd's Rules.

Description of Engine.	Diameter of Intermediate Shaft in Inches.
Compound — Two cranks at right angles,	$(\cdot 04\ A + \cdot 006\ D + \cdot 02\ S) \times \sqrt[3]{P}$.
Triple—Three cranks at equal angles,	$(\cdot 038\ A + \cdot 009\ B + \cdot 002\ D + \cdot 0165\ S) \times \sqrt[3]{P}$.
Quadruple — Two cranks at right angles,	$(\cdot 034\ A + \cdot 011\ B + \cdot 004\ C + \cdot 0014\ D + \cdot 016\ S) \times \sqrt[3]{P}$.
Quadruple—Three cranks,	$(\cdot 028\ A + \cdot 014\ B + \cdot 006\ C + \cdot 0017\ D + \cdot 015\ S) \times \sqrt[3]{P}$.
Do. —Four cranks,	$(\cdot 033\ A + \cdot 01\ B + \cdot 004\ C + \cdot 0013\ D + \cdot 0155\ S) \times \sqrt[3]{P}$.

Where A is diameter of H.P. cylinder in inches.
B „ first I.P. „
C „ second I.P. „
D „ L.P. „
S is stroke of pistons in inches.
P is boiler pressure above atmosphere in lbs. per square inch.

The diameter of the crank-shaft and the *thrust-shaft* between the collars to be at least $\frac{21}{20}$ of that of the intermediate shafts. The diameter of thrust-shaft may be tapered off at each end to the same size as that of the intermediate shafts.

The diameter of the screw-shaft to be equal to the diameter of intermediate shaft multiplied by $\left(\cdot 63 + \frac{\cdot 03\ P}{T}\right)$, where P is the diameter of the propeller and T that of the intermediate shaft, both in inches. In no case, however, must the diameter of the screw-shaft be less than 1·07 T.

Note.—This size of screw-shaft is intended to apply when continuous brass liners are fitted the whole length of stern tube. If no liners are used, or if two separate ones, then the shaft must be $\frac{21}{20}$ of that given by the Rule.

Lloyd's Register Rules for Oil Engine Shafts.

The diameter of the shafts of oil engines for driving screw propellers may be determined in the following manner when made of ordinary mild steel and the maximum pressure in cylinders does not exceed 500 lbs. per square inch :—

(1) For petrol or paraffin engines for smooth-water service, *diameter of crank-shaft* in inches $= C\sqrt[3]{D^2 \times S}$. D is the diameter, and S the stroke in inches. $C = 0{\cdot}34$; for engines with six cylinders $C = 0{\cdot}36$; if for open sea work add 0·02 to each.

(2) Single-acting Diesel type engines :—

$$\text{Diameter of crank shaft} = \sqrt[3]{D^2 \times (A\,S + B\,L)}.$$

Here L is the distance apart of bearings (inner edges); the values of (A S+B L) are as in this table.

TABLE XXXVII.—CRANK COEFFICIENTS.

	Four-cycle Single-acting Engine.	Two-cycle Single-acting Engine.	Value of Coefficient, AS + BL.
a	4 or 6 cylinders.	2 to 3 cylinders.	·089 S + ·056 L
b	8 ,,	4 ,,	·099 S + ·054 L
c	10 to 12 ,,	5 to 6 ,,	·111 S + ·052 L
d	16 ,,	8 ,,	·131 S + ·050 L

For auxiliary engines values 5 per cent. less.

In solid forged shafts, breadth of webs 1·33 diameter, and thickness 0·56 diameter of shaft or of equivalent strength.

(3) When no flywheel is fitted—

$$\text{Diameter of intermediate shafts} = \text{coefficient} \times \sqrt[3]{D^2 \times S}.$$

The value of this coefficient is as follows :—

TABLE XXXVIIA.

Four-cycle Single-acting.	Two-cycle Single-acting.	Value of Coefficient.
4 cylinders.	2 cylinders.	0·456
6, 8, 10, or 12 cycles.	3, 4, 5, or 6 cylinders.	0·436
16 cycles.	8 cylinders.	0·466

N.B.—When the stroke is not less than 1·2 nor more than 1·6 diameter of cylinder, then

$$\text{Diameter of internal shaft} = \text{coefficient } (\cdot 735\,D + \cdot 273\,S).$$

(4) When flywheels are fitted, then coefficient :—

TABLE XXXVIIB.

Four-cylinder Single-acting Engine.	Two-cycle Single-acting Engine.	Value of Coefficient.	Four-cylinder Single.acting Engine.	Two-cycle Single-acting Engine.	Value of Coefficient.
4 cylinders.	2 cylinders.	0·405	10 cylinders.	5 cylinders.	0·420
6 ,,	3 ,,	0·400	12 ,,	6 ,,	0·427
8 ,,	4 ,,	0·409	16 ,,	8 ,,	0·461

(5) Diameter of screw shaft $= T\left(0{\cdot}63 + \frac{0{\cdot}03\,P}{T}\right)$, and never less than 1·07 T; T being diameter of intermediate shaft and D that of propeller in inches.

The Board of Trade Rules for Shafting of Oil Engines is the same as the above rules of Lloyd's Register.

THE BRITISH CORPORATION RULES FOR SHAFTS.

Diameter of Shafting.—The minimum diameters of crank, thrust, propeller, and intermediate shafts may be found from the following formulæ, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration :—

$$D = \sqrt[3]{\frac{P \times L^2 \times S}{C}} \times B.$$

Where D = diameter of shaft.
P = absolute pressure—*i.e.*, boiler pressure + 15 lbs.
S = stroke of engine, in inches.
L = diameter of low-pressure cylinder, in inches.
B = 1·0 for crank- and thrust-shafts.
B = 0·95 for intermediate shafts.
B = for propeller shafts to be taken from the following Table :—

TABLE XXXVIII.—British Corporation Factors for Shafts.

Coefficient of Displacement of Vessel at four-fifths Moulded Depth.	Ratio of Diameter of Propeller to Diameter of Crank-shaft.					
	13	14	15	16	17	18
·6	1·0	1·01	1·02	1·03	1·04	1·05
·62	1·01	1·02	1·03	1·04	1·05	1·06
·64	1·02	1·03	1·04	1·05	1·06	1·07
·66	1·03	1·04	1·05	1·06	1·07	1·08
·68	1·04	1·05	1·06	1·07	1·08	1·09
·70	1·05	1·06	1·07	1·08	1·09	1·10
·72	1·06	1·07	1·08	1·09	1·10	1·11
·74	1·07	1·08	1·09	1·10	1·11	1·12
·76	1·08	1·09	1·10	1·11	1·12	1·13
·78	1·09	1·10	1·11	1·12	1·13	1·14
·80	1·10	1·11	1·12	1·13	1·14	1·15

The value of the divisor C in the formula depends on the ratio $\frac{L^2}{H^2}$, where L = diameter of low-pressure cylinder and H of high-pressure cylinder, in inches:—

$\frac{L^2}{H^2}$	Two Cranks at 90°, Compound or Quadruple, also Three Cranks at 120°, Quadruple Expansion.	Three Cranks at 120°, Triple Expansion.	Four Cranks at 90°, Quadruple Expansion.
Ratio 3	9,910	...	...
,, 3⅛	10,160	...	...
,, 3¼	10,410	...	...
,, 3⅜	10,660	...	...
,, 3½	10,910	...	...
,, 3⅝	11,160	...	...
,, 3¾	11,410	...	...
,, 3⅞	11,660	...	...
,, 4	11,910	...	...
,, 4⅛	12,160	...	...
,, 4¼	12,410	...	...
,, 4⅜	12,660	...	...
,, 4½	12,910	13,650	...
,, 4¾	13,375	14,160	...
,, 5	13,840	14,670	...
,, 5¼	14,305	15,180	...
,, 5½	14,770	15,690	...
,, 5¾	15,235	16,200	...
,, 6	15,700	16,710	...
,, 6½	16,630	17,730	...
,, 7	17,560	18,630	...
,, 7½	18,410	19,530	...
,, 8	19,260	20,430	22,660
,, 8½	20,110	21,330	23,660
,, 9	20,960	22,200	24,660
,, 9½	21,750	23,070	25,660
,, 10	22,540	23,940	26,580
,, 10½	23,330	24,810	27,500
,, 11	24,120	25,660	28,420
,, 11½	24,900	26,500	29,340
,, 12	25,680	27,340	30,260

Rules of the Bureau Veritas.

Shafts for Screw Steamers.

(*a*) *Crank Shafts.*—§ 7. When the crank of a screw engine is not overhung, the diameter of the shaft shall be determined by one of the following formulæ :—

For non-compound condensing engines :

$$d = \sqrt[3]{\frac{n\,P\,L\,D^2}{C}}. \quad - \quad - \quad - \quad - \quad (A)$$

For double, triple, and quadruple-expansion engines :

$$d = \sqrt[3]{\frac{P\,L\,(n_1\,D_1^2 + 0{\cdot}1\,n\,D^2)}{C}}. \quad - \quad (B)$$

For shafts having a single overhung crank, the form under the radical sign is to be multiplied by

$$s + \sqrt{s^2 + 1}.$$

For two-cylinder single-crank tandem engines the formula will therefore be :

$$d = \sqrt[3]{\frac{P\,L\,(D_1^2 + 0{\cdot}1\,D^2)\,(s + \sqrt{s^2 + 1})}{C}}. \quad (C)$$

In those formulæ :

d = diameter of the after shaft bearing in inches.

n_1 = number of high-pressure cylinders.

D_1 = diameter of each high-pressure cylinder in inches. If there are several high-pressure cylinders the diameters of which are not the same, $n_1\,D_1^2$ represents the sum of the squares of their respective diameters.

n = number of low-pressure cylinders.

D = diameter of each low-pressure cylinder in inches. If there are several low-pressure cylinders the diameters of which are not the same, $n\,D^2$ represents the sum of the squares of their respective diameters.

N. B.—For triple or quadruple-expansion engines the intermediate cylinders do not come into account in the formulæ.

L = length of stroke in inches, common to all pistons.

P = boiler pressure above atmosphere in pounds per square inch.

$s = \frac{a}{r}$ (see fig. 110*a*). In order to determine a, B is supposed to be situated half-way the length of the bearing, unless the latter be longer than $1\frac{1}{2}$ times the diameter ; in this case B C may be considered as being equal to $\frac{3}{4}$ of the diameter.

C = a constant, the values of which are given below for certain cases. The values given apply to navigation in a sea-way ; for smooth water (tugs excepted) the constants may be increased by 30 per cent.

If it is above 15 inches, it should be increased by an amount to be determined by the Administration ; for built-up shafts, however, this latter increase will not be required.

For hollow shafts the diameter must be increased by

1 per cent. if the diameter of the hole is 0·4 of the outside diameter.					
2	,,	,,	,,	0·5	,,
5	,,	,,	,,	0·6	,,
10	,,	,,	,,	0·7	,,

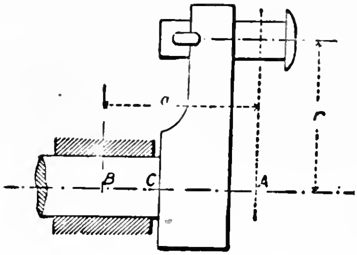


Fig. 110*a*.

If the hole is under 0·4 of the outside diameter, no increase will be required.

The Administration may allow a reduction on the diameter in certain special cases, for instance in well-balanced engines with light moving parts, or for very superior workmanship, etc. On the other hand, the Administration may require an augmentation for engines which differ much from the average proportions found in practice, thus, for instance, for engines having a comparatively small stroke; for compound engines, the low-pressure cylinder of which has a very large size compared with the high-pressure cylinder, etc.

Values of Constant in Formula (A) is 6,230; in (B) for double-expansion two-cranks 90° it is 3,400; for triple three-cranks 120°, 3,900; for quadruples and triples four-cranks 90°, 4,000, and for special crank setting for minimum torsion add 100.

Other Cases.

(*b*) *Propeller, Tunnel, and Thrust Shafts.*—The diameter of the crank-shaft must be $\left(1{\cdot}7\,\frac{D}{d} - 15\right)$ per cent. in excess of the diameter of the crank-shaft calculated from one of the formulæ (A), (B), or (C); D being the diameter of the propeller in inches, and d the diameter of the crank-shaft in inches. It is recommended to fit the propeller shaft in such a way that it cannot move endways, if for some reason or other it has been uncoupled from the rest of the shafting. Liners fitted on propeller shafts to be tapered off at ends.

For tunnel-shafts a reduction of 6 per cent. on the diameter of the crank-shaft will be allowed.

The diameter of thrust shaft at the bottom of the collars, both between and immediately beyond these latter, to be equal to that of the crank-shaft, and tapered off at each end to the smaller diameter of the body of the shaft. The thrust of the screw propeller must be taken up by an efficient thrust block, so as to prevent any fore and aft strain on the crank-shaft.

Shafts for Paddle Steamers.

§ 8. In side wheel steamers having double, triple, or quadruple-expansion engines with an intermediate shaft, each end of which carries an overhung crank-pin fitting loosely into an eye of the paddle-shaft crank, the bearing of

TABLE XXXVIIIa.—Horse-Power transmissible through Shafts by Turbines or Electric Motors in accord with B. of T. Rules.

N.B.—If by Two-Crank Compound multiply by 0·7, by Three-Cranks 0·778, by Four-Cranks 0·746.

Diameter of Shaft.	Revolutions per Minute.														
	75	100	125	150	175	200	250	300	350	400	450	500	600	700	800
6·0	269	358	448	537	627	717	896	1,075	1,254	1,433	1,612	1,791	2,149	2,508	2,866
6·5	326	434	578	723	868	1,012	1,265	1,518	1,772	2,025	2,278	2,531	3,037	3,543	4,050
7·0	427	569	711	853	996	1,138	1,422	1,707	1,991	2,276	2,560	2,845	3,414	3,982	4,551
7·5	524	700	875	1,050	1,225	1,400	1,749	2,099	2,449	2,799	3,149	3,499	4,198	4,898	5,598
8·0	647	863	1,078	1,294	1,509	1,725	2,156	2,587	3,019	3,450	3,881	4,313	5,175	6,038	6,900
8·5	764	1,018	1,273	1,527	1,782	2,037	2,545	3,054	3,563	4,173	4,682	5,191	6,229	7,267	8,305
9·0	907	1,209	1,511	1,814	2,114	2,414	3,017	3,619	4,222	4,825	5,427	6,030	7,236	8,442	9,648
9·5	1,067	1,422	1,778	2,155	2,489	2,844	3,555	4,266	4,977	5,688	6,399	7,110	8,532	9,954	11,376
10	1,244	1,659	2,074	2,489	2,904	3,319	4,148	4,977	5,806	6,635	7,464	8,293	9,952	11,610	13,270
11	1,656	2,207	2,760	3,311	3,863	4,415	5,519	6,623	7,727	8,831	9,935	11,040	13,240	..	..
12	2,150	2,867	3,584	4,301	5,018	5,735	7,168	8,601	10,034	11,470	12,900	14,330	..	..	..
13	2,733	3,644	4,555	5,466	6,377	7,288	9,110	10,930	12,750	14,570	16,400	..	..	..	..
14	3,414	4,552	5,690	6,828	7,966	9,104	11,380	13,650	15,930	18,200	..	..	..	..	..
15	4,198	5,597	6,996	8,395	9,794	11,193	13.991	16,790	19,580	..	..	..	..	..	..
16	5,100	6,800	8,500	10,200	11,900	13,600	17,000	..	..	..	..	..	..	..	..
17	6,112	8,149	10,186	12,233	14,260	16,297	20,370	..	..	..	..	..	..	..	..
18	7,255	9,673	12,091	14,509	16,907	19,345	..	..	..	..	..	..	..	..	..
19	8,532	11,376	14,220	17,064	19,908	22,752	..	..	..	..	..	..	..	..	..
20	9,952	13,269	16,586	19,903	23,220	26,537	..	..	..	..	..	..	..	..	..
21	11,770	15,690	19,600	23,540	..	..	..	..	..	..	..	..	..	..	..
22	13,240	17,660	22,070	26,500	..	..	..	..	..	..	..	..	..	..	..

the latter (see A of the following sketch) must have its diameter calculated from the formula:

$$d = \sqrt[3]{\frac{\mathrm{P\,L}\,(n_1\,\mathrm{D}_1^2 + 0{\cdot}1\,n\,\mathrm{D}^2)\,(s + \sqrt{s^2+1})}{\mathrm{C}}} \quad . \quad . \quad (\mathrm{D})$$

where the letters have the same meaning as before (§ 7), except that a, for determining $s = \frac{a}{r}$, is to be measured as shown in the sketch below, the point B being the middle of the bearing.

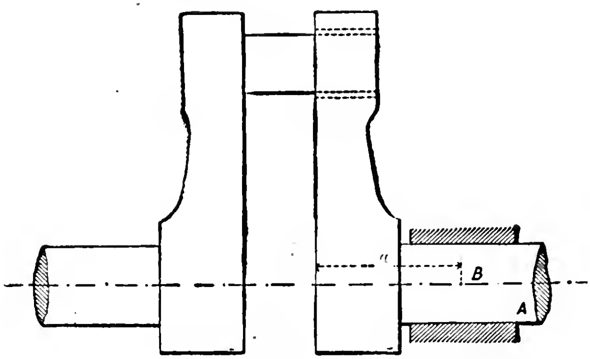


Fig. 110*b*.

For two-cylinder compound receiver engines with two cranks at 90°.

C = 13,000 for navigation in smooth water.
7,100 for coasting vessels.
5,700 for sea-going vessels.

For triple-expansion engines with three cylinders and cranks at 120°.

C = 14,900 for navigation in smooth water.
8,150 for coasting vessels.
6,540 for sea-going vessels.

The diameter of the outer bearing of the paddle shaft and of the intermediate shaft to be submitted to the Administration or the Surveyors for approval. The same applies to other cases not dealt with in this paragraph.

Turbine Shafting (*Bureau Veritas*).

$$\text{Diameter of tunnel-shafts} = \sqrt[3]{70 \times \frac{\mathrm{H.P.}}{\mathrm{R}}} = d_t$$

H.P. is the shaft horse-power, R the corresponding revolutions per minute.

The rotor shaft diameter $= 1{\cdot}05 \times d_t$ at least.

$$\text{Diameter of screw-shaft} = d_t + \frac{\text{diameter of screw in inches}}{160}.$$

Shafts of Oil Engines (*Bureau Veritas*).

$$\text{Diameter of crank shaft} = C\sqrt[3]{D^2 \times S},$$

where D is the diameter of cylinder, and S the stroke of piston in inches. C is a factor which varies inversely as $\frac{S}{D}$, and is 0·493 when $\frac{S}{D}$ is 2·0, and 0·560 when it is only 0·85. $\frac{S}{D}$ is frequently 1·5. Then C = 0·520.

Steel Shafts (*Bureau Veritas*).

§ 9. Shafts made of steel must have a tensile strength of 26 to 30½ tons, and test pieces cut from the forging must satisfactorily withstand the prescribed tests. When it is desired to make use of steel of softer or harder quality, the corresponding alteration in the diameter must be submitted to the Administration (of the Bureau Veritas) for approval.

Summary.—The following is a summary of certain parts of this Chapter which will be useful for reference :—

$$\text{Rule 1.—Diameter of shaft} = \sqrt[3]{\frac{\text{I.H.P.}}{\text{revolutions}}} \times F.$$

The various values of F can be ascertained from the following table, where p is the absolute initial pressure, or that at which the safety valves are loaded + 15 lbs., when there is no reducing valve :—

TABLE XXXIX.—Values of Factor F in Formula for Shafts

$$d = \sqrt[3]{\frac{\text{I.H.P.}}{R}} \times F.$$

	F for Crank Shaft.	F for Tunnel Shaft.
Single-crank expansive 1 cyl. Cut-off, 0.2 stroke,	$170 + 3\sqrt{p}$	$135 + 3\sqrt{p}$
,, ,, 1 ,, ,, 0·6 ,,	$113 + 3\sqrt{p}$	$85 + 3\sqrt{p}$
,, compound 2 ,, ,, 0·6 ,,	$100 + 3\sqrt{p}$	$75 + 3\sqrt{p}$
Two-crank 90° expansive 2 ,, ,, 0·2 ,,	$86 + 3\sqrt{p}$	$70 + 3\sqrt{p}$
,, ,, 2 ,, ,, 0·6 ,,	$68 + 3\sqrt{p}$	$55 + 3\sqrt{p}$
,, compound 2 ,, ,, 0·6 ,,	$63 + 3\sqrt{p}$	$50 + 3\sqrt{p}$
,, quadruple 4 ,, ,, 0·6 ,,	$58 + 3\sqrt{p}$	$45 + 3\sqrt{p}$
Three-crank 120° compound 3 ,, ,, 0·5 ,,	$51 + 3\sqrt{p}$	$40 + 3\sqrt{p}$
,, triple 3 ,, ,, 0·6 ,,	$41 + 3\sqrt{p}$	$30 + 3\sqrt{p}$
Four-crank 90° triple 4 ,, ,, 0·6 ,,	$47 + 3\sqrt{p}$	$35 + 3\sqrt{p}$
,, quadruple 4 ,, ,, 0·6 ,,	$55 + 3\sqrt{p}$	$38 + 3\sqrt{p}$
Three-crank triple-compound naval engines (hollow) .	72	63
Four-crank ,, ,, ,, (,,) .	75	61
Turbines taking S.H.P., ...	..	56

The Diameter of the Propeller Shaft of a screw ship can be obtained with a close approximation to the correct size by adding to the diameter of the tunnel-shaft an allowance based on the diameter of the screw and the conditions of working. If d be the diameter of the screw-shaft at the outer bush, d_1 the diameter of the tunnel-shaft or that to resist torque only, both in inches, and D the diameter of the screw in feet. Then

$$d = (d_1 + x\,\mathrm{D}) \text{ inches.}$$

The value of x for a single screw ship working in smooth water only is 0·06.

For ocean-going ships with single screws, . . . $x = 0{\cdot}075$.
For twin-screw ships with outer brackets, . . . $x = 0{\cdot}09$.

For Example.—An ocean-going single-screw ship has a propeller 18 feet diameter, and the tunnel-shafts are 13 inches diameter, what should be that of the propeller shaft.

$$\text{Diameter of propeller-shaft} = 13 + 0{\cdot}075 \times 18 = 14{\cdot}35 \text{ inches.}$$

Example.—A twin-screw steamer having stern brackets and a screw 12 feet diameter, has tunnel-shafts $11\frac{1}{4}$ inches diameter, what size should be the outer propeller-shaft?

$$\text{Diameter shaft} = 11{\cdot}25 + 0{\cdot}09 \times 12 = 12{\cdot}33.$$

The crank- and screw-shafts of ships which are run at full speed only occasionally, and for short periods, may be calculated by taking F at half the above values.

Example.—To find the diameter of the crank-shaft for a three-crank triple-compound engine having cylinders 30 inches, 45 inches, and 75 inches diameter, and a stroke of 50 inches. The load on safety valve is 165 lbs., the revolutions 90, and the I.H.P. 3,300.

(1) By above Rough Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{3300}{90} \times 41} + 3\sqrt{180} = 14{\cdot}39 \text{ ins.}$$

(2) By Board of Trade Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{25 \times 180 \times 75^2}{1110\left(2 + \frac{75^2}{30^2}\right)}} = 14{\cdot}12 \text{ ins.}$$

(3) By Lloyd's Rules.

Diameter of crank-shaft

$$= \frac{21}{20}(\cdot 038 \times 30 + \cdot 009 \times 45 + \cdot 002 \times 75 + \cdot 0165 \times 50)\sqrt[3]{165} = 14{\cdot}5 \text{ ins.}$$

(4) By British Corporation Rule.

$$\text{Diameter of crank-shaft} = \sqrt[3]{\frac{180 \times 75^2 \times 50}{17{,}300}} = 14{\cdot}3 \text{ ins.}$$

(5) By the Rules of the Bureau Veritas.

$$\text{Diameter of shaft} = \sqrt[3]{\frac{165 \times 50\,(30^2 + 0{\cdot}1 \times 75^2)}{3900}} = 14{\cdot}56 \text{ ins.}$$

TABLE XXXIXa.—Shafts for Paddle Engines.

Description of Engine.	Value of F Intermediate Shaft Journal.	Value of F Paddle-shaft Inner Journal.	Value of F Paddle-shaft Outer Journal.
Single-crank single-cylinder, - -	...	80	100
Two-crank two-cylinder; cranks virtually at right angles, and connected by link, - - - - -	...	58	65
Two-crank two-cylinder, with intermediate shaft; cranks at right angles, - - - - - - -	58	50	65
Two-crank two-cylinder; solid crank-shaft; cranks at right angles, -	...	55	65

For paddle steamers working only in smooth water the above values of F may be reduced 20 per cent.

The above values of F were for iron shafts, but it is better not to reduce them when employing mild steel, unless the forgings are from ingots of the highest quality.

Details of Crank-shafts.

Crank-arms if forged solid with the shaft—

*Breadth - - - - = 1·1 × diameter of shaft.
Thickness - - - = 0·75 × „
Diameter of coupling - = 2·0 × „
Thickness „ - - = 0·27 × „
Number of coupling bolts = 2 + diameter of shaft in inches ÷ 2.

Diameter „ „ = diameter of shaft ÷ $\frac{4n + 19}{10}$.

Diameter of crank-pins - = 1 to 1·1 × diameter of shaft.
Length „ - = 1 to $1\frac{1}{2}$ the diameter of shaft.
Length of journals - - = 1 to $1\frac{1}{2}$ „

The following convenient rules give sizes closely approximating to those found in practice, and may be used to obtain the diameter of the shafts, preliminary to making a more elaborate calculation. They are, therefore, very useful in the initial stages of a marine engine design to enable the designer to get on with the work; but being purely empirical they should be used with some caution :—

d is the diameter of the cylinder in inches.
d_m „ „ M.P. „
D „ „ L.P. „
S the stroke also in inches.
F a factor, which for the crank-shaft of ordinary compound engines with cranks at 90° is 12; and for the tunnel-shafts is 13; for three-crank triple-compound engines F is 14 for the crank-shaft and 15·2 for the tunnel-shafts; four cranks, 13 and 14·2.

* Breadth² × thickness = 0·9 × d^2.

(1) Ordinary compound engines

$$\text{Diameter of shaft} = \frac{d + D + S}{F}.$$

(2) Triple-compound three-crank engines

$$\text{Diameter of shaft} = \frac{d + d_m + D + S}{F}.$$

Taking the same example as before

$$\text{Diameter of crank-shaft} = \frac{30 + 45 + 75 + 50}{14} = 14{\cdot}3 \text{ inches.}$$

Rule for Determining quickly the Diameter of Crank-shaft.—Taking D as the diameter of each L.P. cylinder of a compound system, and S the stroke both in inches, and p the initial pressure absolute. Then the diameter of the crank-shaft may be found rapidly and with close approximation to correctness by the following:—

TABLE XXXIX*b*.—Crank-shafts of Screw Engines.

Description of Engine.	Diameter. Crank-shaft.
Two cranks at 90° compound, two cylinders, ...	$\frac{1{\cdot}6\, D + S}{110} \sqrt{p}$
Three cranks at 120° compound, three cylinders, . . .	$\frac{2{\cdot}85\, D + S}{150} \sqrt{p}$
Three cranks at 120° triple-compound, three cylinders, . .	$\frac{2\, D + S}{185} \sqrt{p}$
Four cranks at 90° triple-compound, four cylinders, . . .	$\frac{3\, D + S}{205} \sqrt{p}$
Four cranks at 90° quadruple-compound, four cylinders, . .	$\frac{2{\cdot}5\, D + S}{225} \sqrt{p}$

The tunnel-shafts should be 0·95 × diameter of crank-shaft.

Board of Trade Rule for Shafts of Turbine Ships is as follows:—

$$\text{Diameter} = \sqrt[3]{\frac{\text{I.H.P.}}{R} \times 40{\cdot}2 \times C},$$

where R is the number of revolutions per minute.

For the intermediate or tunnel-shafts, ... C = 1·500.
,, propeller-shafts, ... C = 1·667.

Hollow Shafts, now so extensively used in special ships, are especially suited for the propeller-shafts of multiple-screw ships, inasmuch as they are less liable to sag under their own weight, and, therefore, freer from the tendency to whirl than solid ones would be under the same circumstances.

If the diameter of the hollow shaft is d, and that of its bore d_1, then

The diameter of the solid shaft of equal strength $d_s = \sqrt[3]{\dfrac{d^4 - d_1^4}{d}}$.

The relation of the weights of these shafts will be $\dfrac{d_s^2}{d^2 - d_1^2}$.

In practice d_1 is about a half of d; assuming this to be so,

$$\text{Ratio of weight of solid to hollow} = \frac{d_s^2}{0{\cdot}75\, d^2}.$$

$$d_s = \sqrt[3]{d^3(1 - 0{\cdot}0625)} = 0{\cdot}979\, d.$$

Substituting this value of d_s,

$$\text{Ratio of weight of solid to hollow} = \frac{958}{750}, \text{ or } 1{\cdot}278.$$

That is, the solid shaft of the same strength is nearly 28 per cent. heavier.

The Outboard Shafts of large naval ships with twin screws and ordinary brackets are of large diameter (as much as 28 inches), with a bore-hole 0·7 of the external diameter. Under these circumstances—

(*a*) $$d_s = \sqrt[3]{\frac{d^4 - (0{\cdot}7d)^4}{d}} = 0{\cdot}91d.$$

(*b*) The ratio of weights $= \dfrac{(0{\cdot}91d)^2}{d^2 - (0{\cdot}7d)^2} = \dfrac{{\cdot}828}{{\cdot}510}$.

That is, the solid shaft is 1·6 times the weight of the hollow one; but in salt water the difference is greater. Taking the specific gravity of steel as 7·86 and salt water as 1·03, then—

(*c*) Hollow shaft $= d^2(1 - 0{\cdot}49) \times 7{\cdot}86 - 1{\cdot}03d^2 = 2{\cdot}98$.

Solid shaft $= (0{\cdot}91d)^2 \times (7{\cdot}86 - 1{\cdot}03) = 5{\cdot}66$.

The ratio is now 1·9.

CHAPTER XIII.

FOUNDATIONS, BED-PLATES, COLUMNS, GUIDES, AND FRAMING.

For the good working of an engine it is essential that the fixed parts, such as bed-plates, framing, etc., shall not only be strong enough to resist the strains to which they are subject, but rigid and stiff enough to prevent any tendency to rack or change of form which would throw abnormal stresses on the working parts. With this object in view it is usual to construct such parts of cast iron or cast steel; and from their form and general construction thus enables the engineer to manufacture a cheap structure having the necessary qualities. But since ordinary cast-iron has not a high tensile strength, and is comparatively unsuited to withstand sudden shocks, structures made of it cannot be so light as if made of wrought-iron or steel; so that when extreme lightness of machinery is aimed at, the framing is usually made of wrought steel, and rigidity given to it by cross bracing, etc. This latter system is, of course, an expensive one in most engines, and only adopted when economy of weight is of more importance than economy of manufacture. Although cast-iron framing and bed-plates are undoubtedly cheaper and better for engines generally, and continue to be used for all sizes in the mercantile marine, a system of construction with wrought iron or steel for very large engines may be adopted with advantage, and taking into account cost of patterns and risk in casting may not be less economical.

Steel manufacturers can now, however, produce large and moderately complicated castings in steel, and the foundations and frames of express engines of large sizes are being made wholly of that material. Increased experience in the making of steel castings has no doubt given greater faith in their soundness as well as permitted of a lower price, and with improvement will be more freely used for columns and foundations (fig. 32). At present they can only be used to advantage in large engines owing to the considerable thickness still demanded by the steel moulders.

Bed-plates and Foundations.—Vertical engines are usually built on a structure called by these names, but are sometimes known as the *sole-plate*. It contains the main bearings for the crank-shaft, and on it are the facings for the columns, condenser, etc., and it used sometimes to contain the waterways leading from the condenser to the pumps. The foundation generally contains only the main bearings, and has facings for the front columns only when it is bolted to feet on the front of the condenser, so that with the latter it forms the engine base. The condenser is in this case lower down, and the weight and cost of half the sole-plate is saved. The part of the foundation fitting to the feet on the condenser front should be of good depth, the flanges strong, notched one into the other, and strongly bolted at top and bottom. The transverse parts of the bed-plate, into which the main bearing brasses are fitted, are sometimes formed like inverted bowstring girders, and are unsupported by the bed built in the ship, but span the space between the fore and aft bearers. This is convenient sometimes, especially when the shaft must be low down in ships having a good rise of floor, and also for very small engines; but it must be of strong form, and as it depends for strength only on the connection to the longitudinal parts of the foundation, it is a most convenient form when steel castings are used, as also for very light engines; additional support may be, and often is, given in the case of large engines by bolting to the ship at the middle (fig. 114). When this particular style is adopted great care should be exercised in designing the foundation, so that the transverse portions have a good extended connection to the longitudinal ones, especially in the direction of the column bases. If the longitudinal parts are flat and straight on the bottom, so as to be in the same plane as the rest of the foundation, they may be formed with flanges and bolted to the steel seating in the ship, and from it receive support and strength.

Main Bearings.—The bearings in which the shaft journals run should approximate, as far as possible, to a hole through a solid support. If it were practicable a hole with a bush of suitable metal in it would form the best possible bearing for a shaft; but since the bearing, however well designed and made, is liable, in course of time, to wear somewhat, it becomes a necessity that there shall be some means of adjusting the brasses, so as to prevent the shaft from having too much movement when they are worn.* In the case of the vertical engine, the weight of the shaft, and the pressure from the piston, act very nearly in the same direction, so that the wear is only vertically above and below the shaft; consequently the adjustment is necessary only in a vertical direction. The greatest loads on the bearings, however, are during the first half of the stroke, and consequently the position of mean pressure on the journals is not exactly vertical; this is also somewhat modified on the upstroke by the tendency of the shaft to roll on the surface of the brasses, and on the downstroke it is aggravated from the same cause. In fitting the brasses for a vertical engine, this should be borne in mind, and every allowance made for taking the wear due to these causes. It is of the utmost importance for the good working and endurance of a crank-shaft that the bearings be rigid in themselves, and that the framework containing them shall be rigid enough to sustain them perfectly in line one with another. Crank-shafts are more severely tried by the giving or springing of the bearings than any other cause, and they were more often broken from want of rigidity in the bed-plates and seatings, than from the normal strains from the pistons, so that a shaft may be of ample size to bear the twisting and bending moments if properly supported in its bearings, and yet give way after a few weeks' work because it is in a light foundation in a weak ship.

The brasses were usually formed as shown in fig. 111, and carefully bedded into the recesses provided for them in the foundation. At one time it was usual to design them with projecting facings, called chipping strips, to avoid the labour of chipping and filing the whole of the surface; this was, however, found to be highly objectionable as engines increased in size; with the increase of boiler pressure, and consequent increased percussive action due to the high initial loads, such an effect was produced on these strips and the cast-iron surface on which they were borne, that engineers have gradually abandoned the practice; the planing machines also have rendered such a device unnecessary, as it is practically as cheap to fit brasses now to bear

* With forced lubrication and the bearings shut off from dust, etc., there is in practice no wear.

over the whole surface as to do so only on strips. The square bottom brass is objectionable on two grounds; one being that it is impossible to remove it in most engines without lifting the shaft, and the other that when it becomes hot it is invariably distorted, from the variation in thickness of metal, with the ultimate result that it is broken through the middle longitudinally.

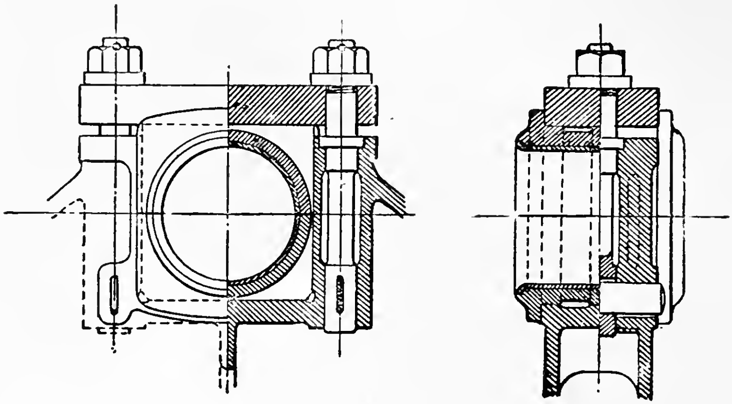


Fig. 111.—Crank-shaft Bearing.

The first of these evils is avoided by making the bottom brass round and of even thickness, so that it can be got out when relieved of the weight of

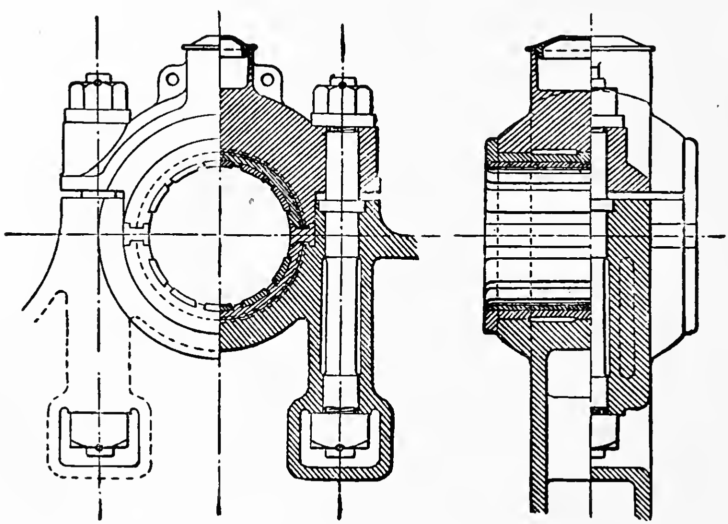


Fig. 112.—Improved Form of Crank-shaft Bearing.

the shaft, by being moved around until it is on the top of the journal. The second evil is also partly avoided by making it of an even thickness; but this form of brass is often found cracked, and is liable to heat from its want of stiffness. Both these brasses, when first heated by abnormal friction, tend to expand along the inner surface, which is in contact with the shaft; this would open the brass, and make the bore of larger diameter, if not resisted by the cooler part near the cast iron, and by the bed-plate itself. If the brass has become hot quickly and excessively, the resistance to expansion produces permanent set on the layers of metal near the journal, consequently on cooling this contracts, the brass closes and tends to grip the shaft; it will then set up sufficient friction to become again hot, and expand sufficiently to ease itself from the shaft, when so long as that temperature is maintained the shaft runs easily in the bearing. This is the reason why some bearings always are a trifle warm, and will not work cool. A continuance of heating and cooling will set up a mechanical action at the structure of the brass, which must end in rupturing it, just as a piece of sheet metal is broken by continually bending backwards and forwards about a certain line.

This pernicious action of the brass can be prevented by securing it to the bed-plate, along its two longitudinal edges, as shown in fig. 112, by an **H**-shaped strip, which holds both top and bottom brasses, so that they cannot move in their beds. This method is a very simple one, and has been found most successful in engines of all sizes.

It is also essential that the bearing to be efficient should be rigid throughout its whole length; this is not the case when the brasses have long overhanging ends, which afford little or no support to the shaft. To this end it is better, when possible, to extend the bed for the brasses, so as to support them over the whole of their length, as shown in fig. 112.

Caps or Keeps for Main Bearings are very generally made of wrought iron, but as stiffness is as necessary as strength, cast steel, as in figs. 113 and 114, or even cast iron, as in fig. 112, may be used with advantage in their construction. A wrought-steel cap, which may be amply strong, is often far from stiff enough, while a cast-iron cap, which is stiff enough for good working, is generally amply strong.

Let d be the diameter of the main-bearing bolts (when there are only two to each cap), t the thickness of the cap, and b its breadth, l is the pitch of the bolts, all in inches; f a factor, which for wrought iron and forged steel is 1, for cast steel 1·2, and for cast iron 2.

$$\text{Thickness of cap} \quad - \quad - = d\sqrt{\frac{l}{b} \times f}.$$

$$\text{Thickness of brass at middle} = \frac{d}{3}\sqrt{\frac{l}{b}}.$$

Main-bearing Bolts.—Each cap is usually held by two bolts, but very large bearings have four bolts, two on each side, so as to avoid large bolts and heavy nuts, and to distribute the load over the cap. When everything is in good order and properly adjusted, the load from the piston should be equally divided between the bolts; but since, from a very slight difference in setting of the nuts, the load may come on three, and sometimes even on two bolts only, due allowance must be made for this. To meet this it should be assumed that each bolt is capable of sustaining one-third the load on the piston. If P is the maximum load on the piston in lbs.,

$$\text{Area of each bolt at bottom of thread} = \frac{P}{3f}.$$

For mild steel f = 5,000 lbs. for small and 7,000 lbs. for large bolts (*v.* Table lxxxiii.).

Also, Diameter of main-bearing bolt = diameter of cylinder $\times \sqrt{\frac{p}{3f}}$.

p is the maximum pressure per square inch, and is, as stated on p. 273, and may be taken generally at 0·75 × absolute boiler pressure for the high-pressure cylinder of a triple engine, to equal the load on the low-pressure piston.

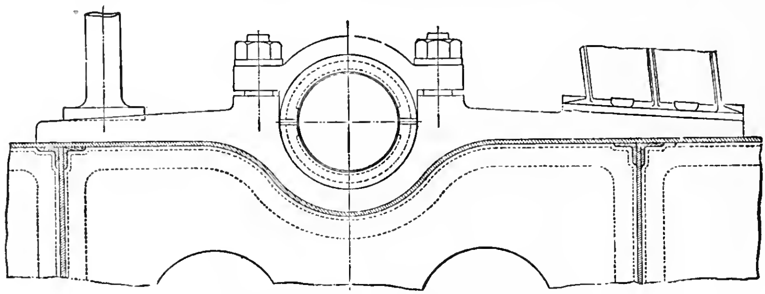


Fig. 113.—Solid Steel Main Bearing Frame.

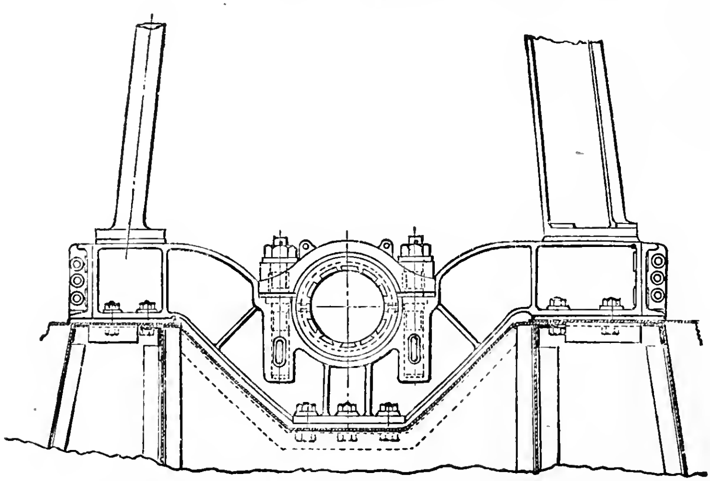


Fig. 114.—Cast-steel Main Bearing Frame for Naval Engines.

Brasses, so called because formerly always made of brass.—They should be made of a metal which itself will withstand wear without wearing the shaft journals, and whose surface is such that the shaft runs on it with minimum amount of friction. The metal must also possess sufficient strength in itself or its shell not to fracture under the percussive loads of the piston, and be free from brittleness, so as not to crack when quickly cooled. Good gun-metal or bronze possessed *all* the qualities essential for brasses when the journal was of wrought iron; but the soft steel as used for shafts will not run satisfactorily on bronze bearings. There are, however, other metals which have certain of these qualities in a higher degree without having them all. Cast iron is harder than ordinary bearing bronze, and when once worn to a smooth surface gives equally good results; but it is liable to fracture from continued shocks and to crack when cooled suddenly. White metals offer least resistance, or produce least friction, but most of them are too soft to be used alone. Of the patent bronzes there are few which are suitable for heavy bearings, and none of them have so far been shown to be much superior to good gun-metal.

White metal carried in a bronze or cast-iron shell is beyond doubt the best bearing for all qualities of steel journals, and if kept well lubricated there is practically no wear on it or the journals.

When a bearing is of ample size, properly designed and constructed, well lubricated and looked after, it may be of almost any of the white metals. If, however, the bearing surface is limited there is a considerable difference in the behaviour of the different metals; but if badly designed and constructed even the best metal will give trouble; moreover, if not properly looked after by the engineer, the best metal and the most careful design are of no avail.

Certain of the white metals have given the best results as a bearing surface, and there is every reason for this, inasmuch as they do not cause abrasion of the shaft, and if their own surface is injured it will not, as a rule, form into fine, hard grit, which grinds both the surfaces, as all the hard bronzes do more or less. When white metal is used it is highly important that the shaft shall bear wholly on it, and not partly on it and partly on the metal containing it, and also that efficient courses for the distribution of the lubricant are provided. For heavy loads the tin compounds are the best.

There are three common methods of fitting the white metal into a setting of other metal—(1) By casting it into oblong recesses; (2) by casting it into a large number of small circular recesses; and (3) by driving strips into longitudinal grooves, in the same way as the lignum vitæ in a stern bush. The last plan is, on the whole, the most satisfactory with large bearings, for the strips are well secured, and extend over the whole length of the bearing, leaving several oil courses longitudinally, and the shaft bears on the white metal only; this also possesses the advantage that a damaged strip may be taken out, and a new one re-fitted with ease, and without heating the brass and running the risk of distorting it. Cast iron is now very often used as a setting for the white metal, and answers the purpose very well indeed, being much harder than brass, and thereby affording the softer metal a better support. When cast iron is used the shell is generally made thicker than when of brass; and sometimes advantage is taken of this to cast the shell hollow, so as to admit of water circulating through it.

The thickness of "brasses" in the crown depends principally on the diameter of journal.

When of bronze	= 0·10 ×	diam. of journal	+ 0·10 in.
,, cast iron	= 0·15 ×	,,	+ 0·15 in.
When fitted with white metal thickness	= 0·20 ×	diameter	+ 0·15 in.

When fitted with white metal strips as follows:—

Thickness of strips . . .	= 0·04 ×	diam. of journal	+ $\frac{1}{8}$ in.
Breadth ,, . . .	= 0·16	,,	+ $\frac{1}{2}$,,
Space between strips . . .	= thickness of strip.		
Thickness of metal beyond . .	= 0·065 ×	diam. of journal	if bronze.
,, ,, . .	= 0·12 ×	,,	,, iron.

Columns.—The columns which support the cylinders of a vertical engine are subject to alternate tensile and compressive stresses from the steam pressure acting on the ends and cover of the cylinder; to a steady compressive stress from the weight of the cylinders, etc.; and to cross-breaking stresses when the ship is rolling and pitching. As a rule, columns, if designed from considerations of mere strength only, would not be stiff enough for good working. The same reasons, therefore, which decide the using of cast iron for foundations strongly influence most engineers to choose this metal for columns. The front columns are often made, however, of wrought iron or steel, turned smooth (*vide* fig. 114); and all the columns for exceedingly light engines are made of wrought steel, well braced together to prevent vibration (*vide* fig. 34). Since cast iron is so superior to wrought iron or steel for resisting compression, and so inferior to either for resisting tension, a good composite column is formed by fitting a steel tie-bar or bars through a hollow cast-iron column, the latter supporting the cylinder while the former holds it down. The chief objection, however to both this composite column and those of wrought steel for large engines is the difficulty of getting good attachment to the cylinder; and since it must be outside the cylinder the columns are necessarily far apart, and away from the centre line of compound engines. To avoid this difficulty wrought-steel columns are formed with a flange at each end like a shaft-coupling; but even then the load on the cylinder is very much concentrated. Columns when of cast iron or of cast steel (*v.* fig. 32) are made of various shapes, and no rule can be laid down in favour of any particular form.

The columns should be so arranged as to support the cylinders and resist the reaction on the foundation. Some engineers, in thoroughly effecting the latter, completely hide from view the working parts, and make all the bearings, etc., very inaccessible (*v.* fig. 38). They should be so placed at the cylinder bottoms that the piston-rod axis is within the lines drawn through the extreme points of back and front columns; and when there are only two columns to each cylinder, the front ones are better to be spread out somewhat, so as to act as struts to resist any tendency to motion of the cylinders when the ship is rolling and pitching, and thereby leave the working parts more open to view. When there are guides on both back and front columns, or when the front columns only have the guides, then this is not possible.

Back columns are generally of different form from the front ones, to suit the guides and bearings for the pump weigh-shaft of the levers when so fitted.

Some engineers have utilised the back columns as exhaust pipes to the condenser, but this is not good practice, inasmuch as the heat of the steam causes them to expand, and when the guides for the piston-rods are on them the heat is conducted to them with prejudicial results; this latter difficulty, however, was sometimes overcome by placing the guides on the front columns. It is also bad practice to expose any important part of the engine structure subject to heavy strain, to unnecessary wear, such as wasting of the inner surfaces of the casting by corrosion.

Guide-plates.—In order to have a sound and hard surface for the piston-rod slides or shoes to work on, the guide-plates should be separate from but secured to the columns; when so fitted they also admit of adjustment, and may be cast hollow, so as to permit of a flow of cooling water through them (*v*. fig. 32). This is especially needful for large quick-running engines, where the speed of piston is very high, and any casual want of lubrication would soon cause most serious damage. Cast iron when once worn smooth provides a splendid surface for a slide, but if by any mischance this surface suffers a little abrasion, it is most difficult to get in good working order again, and cannot be relied on to work well until replaned or ground smooth.

The face of the guide plates should have good oil courses cut on it, so that the lubricant is well distributed, and they should be cut deep enough to retain it and also prevent them being choked with the soapy deposit from some kinds of oil. The piston-rod slide should always be provided with a comb, which will carry the lubricant from the drip-boxes, and spread it over the face of the guide at every stroke; the plate may with advantage have recesses in the upper part to catch the oil so spread, where it is retained and stored to trickle down at leisure.

Framing.—Horizontal engines required a different arrangement of bed-plate and framing from that of the vertical type. Trunk and return connecting-rod engines had no sole-plate proper, as the cylinders were connected to the condenser casting (fig. 31) by **A** frames, which contain the main bearings; the trunk engine required no guides, and those for the crossheads of the return connecting-rod engine were on each side of the condenser. These frames had to be sufficiently strong to take the whole load from the pistons, and stiff enough to be rigid under them, otherwise the crank-shaft was liable to distortion. The usual form approximated to the letter **A**, the two feet being connected to the cylinder front, one at top and one at bottom. in line with the brackets on which the cylinder sat, and by which it was secured to the seatings in the ship. Projecting feet were cast on the cylinder front to meet those of the frames, so that the connection was made with driven bolts.

Usually there were only three frames to a two-cylinder engine, and four frames to a three-cylinder engine, the middle ones being very much stronger than the other two, as they were required to take part of the strain of both engines.

Horizontal direct-acting engines have a sole-plate, which connects the cylinders to the condenser casting, and contains the guides for the piston-rods, and the brackets for the main-bearings; these latter are usually stayed to the cylinder tops by tie-bars through cast-iron struts or by steel tie-rods only.

The framing for the older diagonal paddle-wheel engines was made somewhat on the same principle as that for the horizontal screw-engine, with modifications (*v*. fig. 19) to suit the altered conditions. The main part of these frames must extend from the cylinder to shaft and down again, so as to form a support for the latter, and having guides for the piston-rod crossheads. Intermediate supports or columns connect this main frame to the foundation. It is now usual to make these frames of steel bars (*vide* fig. 20), and for some very light draught steamers of large power frames made of steel sections and plates have been found considerably lighter than the ordinary frames, and can be designed to add materially to the stiffness and strength of the ship in the neighbourhood of the machinery (*v*. fig. 22). A modified form of this kind of frame is imperative, when exceedingly light draught is a necessity, as the hull is so light, to comply with the requirements, as to be unable by itself to stand the racking strains from the engines.

Even when weight is of secondary consideration, the wrought-steel frame is preferable to the cast iron, and when the cost of patterns is taken into account, it is no more expensive. The bearings for the shaft, and the guide-plates for crossheads, are, of course, of cast iron or cast steel fitted to the wrought-steel work direct.

Side stiffness and stability are obtained by connecting the four frames by wrought-iron tie-bars through the top (*v.* fig. 22), and by the main beam before the shaft. The main-bearings for the shaft of a diagonal engine are sometimes so set that the shaft can be lifted vertically; but a better plan is to set them at a slight angle, so that their centre line is in the direction of the *resultant* of the *weight* of shaft, etc., and *mean pressure on the journals* due to the thrust on the connecting-rod.

Entablature of Oscillating and Steeple Engines.—This is usually of cast iron, but may with advantage be made of wrought-steel plates and angles, or, better still, of cast steel, as the strains on it from the overhung cranks are severe and concentrated owing to its being supported at so few points. It was no uncommon thing formerly to find it broken and patched after only a few months' use, and not many work without a certain amount of give-and-take, which must tend to produce rupture in course of time.

It is usually supported (fig. 27) by four columns to each crank, and additional stiffness and stability are imparted by diagonal cross braces to the foundation at each end. It is seldom possible to place the supporting columns in line with the main girders of the entablature, but when possible this should always be done, so as to avoid the canting action which is caused by the centre of support not being in the same plane with the centre of force on the journals. When this is not possible, the sides of the entablature connecting the main girders or rockers should be of extra stiffness, and well connected to them by spreading out webs or fillets. Special advantage should also be taken of the main beams worked into the ship, to form a powerful tie to the entablature girders, and to prevent their tendency to canting. This can be done generally by multiplying the number of the bolts, and fitting cast-iron filling pieces in lieu of hardwood ones only. The bottoms of the cross-bearers should, when possible, be tied together by bars parallel with the crank-shaft to keep them from "giving" or twisting. In the case of steeple engines (fig. 28) the thrust of the connecting-rod on the guides caused great stress on the entablature and framing, and necessitated diagonal ties between it and the foundation. The horizontal force in an oscillating engine also puts cross or horizontal strains on the entablature which the ordinary columns are not fitted to resist, hence the diagonal cross frame which Penn generally fitted.

To resist, as far as possible, the tendency to spring, the supporting columns should be of extra size, with strong and broad flanges.

When there are four supporting columns of wrought steel, their diameter in the body should be 0·7 the diameter of the piston-rod, and at each end 0·55.

The collars or shoulders on which the entablature is supported, should be equal in diameter to that of the piston-rod, and 0·2 the diameter of the piston-rod in thickness.

If the columns happen to come in line, or nearly so, with the centre of the shaft journal, they may be 10 per cent. less in diameter than given above. The breadth of the rockers should be not less than the diameter of the shaft journals, and the depth at the centre should be calculated as for a box-girder, subject to sudden loads applied at the middle of its length, which is measured from column to column.

Roughly speaking, the depth of the rocker under the bearing brass should not be less than four times the diameter of the piston-rod for engines of ordinary dimensions.

The thickness of metal of sides of rockers

$$= 0{\cdot}4 \sqrt{\text{diameter of piston-rod.}}$$

Thickness of top and bottom $= 0{\cdot}6 \sqrt{\text{diameter of piston-rod.}}$

The bottom brasses of the main bearings should be round, so that the recesses for them may tend to strengthen the rockers rather than weaken them, as would be the case if square-bottomed.

CHAPTER XIV.

THE CONDENSER.

The function of the condenser is to cool the steam on leaving the engine, so as to reduce its pressure to a minimum; in doing this the steam is converted back into water. The very early engines could only work by the aid of condensation, as the steam with which they were supplied was generally of a little lower pressure than the atmosphere; it is, in fact, owing to this that the steam-engine owes its birth, for steam was preferred by the early mechanicians because it was so readily changed from vapour to liquid, and so produced that vacuum which Nature was supposed to abhor, and to fill which she would perform the work of horses. The proper relation of the condenser to the engine is better understood by following the early history of the steam-engine from the day when the cooling water was admitted to the cylinder containing steam, and then allowed to run freely away from the bottom on the descent of the piston, to the time when Watt, having perceived the waste of energy in always forcing the piston up against the atmospheric pressure, and in admitting the hot steam into the cold cylinder, made the engine double acting, and effected the condensation in a separate chamber. The jet or spray of water continued long after Watt's time as the means of cooling the steam, and gave in later days the distinguishing name to the condenser, which is now nearly entirely superseded by a more perfect apparatus.

The Common or Jet Condenser, now really *uncommon* on shipboard, consists essentially of an air-tight chamber, into which the steam flows from the cylinder after having been exhausted of its available energy; at its entry the steam is intercepted by a spray of water, caused by the inrush from the sea through small holes or narrow slits in a pipe placed across the steamway. If the spray is fine, like a shower of rain, it mixes mechanically with the steam, as well as cools it by surface contact; should the pipe have slits, so as to cause the water to flow in thin broad streams like ribbons, the cooling is effected principally by surface contact. The result in either case is the turning of the steam into water, which falls to the bottom, and is pumped away by the *air*-pump. It might be supposed that the mere turning of the steam into water, thereby causing it to occupy far less space, will produce the vacuum in the condenser; in practice it does, but to so slight an extent and of such an evanescent nature, that unless some other means were at hand, this condenser would be nearly useless. Water readily absorbs air when freely exposed to the atmosphere, and gives it up again on being heated, or relieved of atmospheric pressure. Fresh feed-water contains air, which becomes mechanically mixed with the steam in the boiler, and passes with it through its various passages, until it enters the condenser, when it parts company with it, and remains as cooled rarified air after the steam is converted to water. The cooling water also contains air, and readily gives most of it up on becoming heated by the exhaust steam, especially under the diminished pressure of the condenser. After a few strokes of the engine a sufficient amount of air will be accumulated to raise the pressure to that of the atmosphere, and although the condenser may be kept quite cool there will be no vacuum, but the reverse. The pump, therefore, which exhausts the condenser draws away the air as well as the water, but since the latter could run away by gravity, it is only the former which is of necessity pumped; hence this pump is rightly named the *air-pump*.

The *shape* of a jet condenser is immaterial so long as the inlet for the steam is high enough to prevent the water running back into the cylinder, and the bottom so shaped that the water will all drain into the air-pump bottom. It was generally formed to suit the ship and the working parts of the engine, and was often a part of the engine framing; the back columns of vertical engines were often utilised for the purpose, and did extremely well, except that occasionally rapid corrosion so weakened them as to become dangerous. The frames of the horizontal engines were arranged by some engineers to do duty for condenser, until the Admiralty finally forbade the practice.

The *capacity* of the jet condenser should not be less than one-fourth that of the cylinder or cylinders exhausting into it, and need not be more than one-half of it, unless the engine is a very quick running one; one-third the capacity is generally, however, sufficient. The objection to a large condenser, beyond its cost and weight, is that a longer time is necessary to form a good vacuum in it; and the objection to a small condenser is its liability to flooding and overflowing to the cylinders, unless the engineer is most attentive.

The amount of injection water depends on the *weight* of steam to be condensed *and its temperature*; the *exact* quantity of water per pound of steam depends on the temperatures of the steam, of the cooling water, and of the "hot-well," or receptacle into which the air-pump delivers the products of the condenser. As the supply of water to the boilers (called the *feed-water*) is taken from the hot-well, and it is an obvious advantage for it to be as warm as possible, the cooling water used is only such as sufficient to produce a good vacuum. With the jet condenser a vacuum of 24 inches was considered fairly good, and 25 inches as much as was possible with most condensers; the temperature corresponding to 24 inches vacuum, or 3 lbs. pressure absolute, is 140°. In actual practice the temperature in the hot-well varied from 110° to 120°, and occasionally as much as 130° was maintained by a careful engineer. To find the quantity of injection water per lb. of steam to be condensed:

Let T_1 be the temperature of the steam whose latent heat is L; T_0 the temperature of the cooling water, whose quantity in lbs. is Q; T_2 the temperature after condensation, or that of the hot-well.

The *total heat* of the steam $= T_1 + L$.

The heat absorbed by the cooling water will be $(T_1 + L) - T_2$.

But the heat absorbed by the cooling water is also represented by $Q\,(T_2 - T_0)$. Hence

$$(T_1 + L) - T_2 = Q\,(T_2 - T_0).$$

Or,

$$Q = (T_1 + L) - T_2 \div (T_2 - T_0).$$

Now $(T_1 + L) - T_2$ is equivalent to the total heat of evaporation from T_2 and at T_1, and is therefore equal to $966° + 0·7 \times 212° + 0·3 \times T_1 - T_2$.

Or, $$Q\,(T_2 - T_0) = 1{,}114° + 0·3 \times T_1 - T_2.$$

Therefore

$$Q = \frac{1{,}114° + 0·3 \times T_1 - T_2}{T_2 - T_0}.$$

Example.—To find the amount of injection water required for an engine, the steam at exhaust being at a pressure of 10 lbs. absolute; the temperature of the sea is 60°, and it is required to keep the hot-well at 120°.

The temperature corresponding to 10 lbs. is 193°.

$$Q = \frac{1{,}114° + 0·3 \times 193° - 120°}{120° - 60°} = 17·53 \text{ lbs.}$$

That is, the amount of injection water is 17·53 times the weight of steam for this particular case.

The allowance made for the injection water of engines working in the temperate zone was usually 27 to 30 times the weight of steam, and for the tropics 30 to 35 times, the summer temperatures being about 60° and 80° F.

The Area of injection orifice and size of pipe is governed by the head of water, vacuum, and length of piping, or, in other words, by the *equivalent head* at the condenser.

Neglecting the resistance to flow at the orifice, and in the pipes and passages, the velocity at the condenser may be found as follows:—

Let h be the head of water above the valve on the condenser in feet, p the pressure in the condenser, and h_1 the equivalent head g for gravity,

$$h_1 = h + (15 - p)\,2·3,$$

and velocity in feet per second $= \sqrt{2\,g\,h_1} = 8·025\,\sqrt{h_1}.$

Example.—To find the theoretical velocity of flow into a condenser in which the vacuum is 26 inches, and the orifice 12 feet below the water-line of the ship. Here

$$h_1 = 12 + (15 - 2)\,2·3 = 42 \text{ feet.}$$

Then

$$\text{velocity} = 8·025\,\sqrt{42} = 52 \text{ feet per second.}$$

In practice owing to loss of head due to resistance at valves in the pipes, etc., the actual velocity was only about half that given by the above rule. Hence, in designing it was usual to calculate on a velocity of only 25 feet per second for shallow draught steamers, and 30 for deeper ones.

From these rules, and with such allowances, the following holds good:—

Area of injection orifice in square inches = number of cubic feet of injection water per minute ÷ 10·4 to 12·5 according to circumstances, or = weight of injection water in pounds per minute ÷ 650 to 780.

A rough rule sometimes used was—

Allow one-fifteenth of a square inch for every cubic foot of water condensed per hour. And another

Area of injection orifice = area of piston ÷ 250.

The injection valve was usually a simple slide or sluice valve, which readily opened or shut, and regulated the amount of water. The handle or lever for working the injection valve must be very near the starting gear, so that the water may be shut off as soon as the engine stops.

Snifting Valve.—It is usual to fit, at the bottom of a jet condenser, a non-return valve, through which the water, etc., may run or be blown out by steam; it shuts by its own weight, and is held on its seat by the pressure of the atmosphere. This is called the *snifting* valve, and it allowed of the condenser being emptied of water and air by steam before starting the engine; it likewise prevents the pressure in the condenser exceeding that of the atmosphere to such an extent as to be dangerous in case of mishap with the injection water or steam, and on that account is a useful and necessary adjunct to the condensers of turbine engines, as there is no hindrance to the full and direct flow of steam into it should the rotor stick. The valve through which the steam is admitted is called the *blow-through valve*, and was a simple mushroom valve, raised by means of a lever, and closed by the steam pressure on the lever being released.

The snifting valve was usually exposed without a casing, so as to be easily inspected or removed in case of being gagged with dirt, etc.; to prevent the water from being spurted about the engine-room, the valve was formed with a curved rim, which completely covered and overhung the seat like an inverted saucer.

The internal injection pipe or *rose* should be placed below the flow of steam, so that the cooling water may pass in a shower twice through the steam.

Surface-Condenser.—It has been seen that with jet condensation the contents of the hot-well consist of a mixture of condensing and condensed water in the proportion of about 30 to 1, so that the water available for feeding the boiler was nearly as salt as sea water. If the cooling water is kept separate from the condensed steam, the latter, which is pure water, may be used as feed-water. The idea was by no means new, since so far back as 1794 Cartwright took out a patent for an engine, in which the steam was condensed on the cold surfaces of two metal cylinders placed one within the other, and having cold water through the inner one and about the outer. An engine was made on this patent, and is said to have given great satisfaction. Brunel, in 1822, took out a patent for the same object; his invention was designed more especially for ships, and consisted of groups of small tubes. In 1833, L. Herbert and J. Don patented an arrangement whereby "the air and steam from the eduction passage is drawn by a fan through or among small tubes, so as to be condensed. The tubes may be below the water." In 1835, W. Symington patented a plan "for condensing the steam from the cylinder, and cooling the surplus water from the air-pump, by tubes laid along the keel exposed to water outside a steam vessel." In 1838, J. B. Humphreys took out a patent for "surface-condensation by leading the steam through tubes in a vessel kept cold by a flow of water." The practical success, however, of this mode of condensation is chiefly due to Mr. Samuel Hall, with whose name the surface-condenser is generally and properly associated. He took out a patent in 1831 for a system of surface-condensation, and in his specification claims, among other things, to condense the waste steam from the safety-valves, and to distil fresh water, to make up loss, by an apparatus, in principle similar to the distillers of to-day. One of the first ships fitted with Hall's condenser and appurtenances was the "Sirius," which made the first voyage under steam from England to America in 1838. In 1837 engines, made for the "Wilberforce," were fitted with Hall's condenser, and it is interesting to note that these engines had piston valves, etc., pretty much as now used. Ignorance and prejudice drove Hall's invention off the market for many years, so that it was not till 1860 that the surface-condenser came into general use, and then only slowly; indeed, it was not until mineral oil was used exclusively for internal lubrication that objections to surface-condensation ceased. In Hall's condensers the steam passed through the tubes and the circulating water outside them; had the cooling water gone through the tubes it is highly probable they would not have "choked with mud and sand" and been condemned, as they were in the case of the "Wilberforce" in 1841.

Condenser Tubes.—It is essential that the material on the surface of which steam is to be condensed should be metallic, thin, and a good conductor of heat, strong enough to resist the pressure of the water on it, amounting to at least 15 lbs. per square inch, and capable of withstanding sudden changes of temperature without fracture and the corrosive action of salt water and distilled water.

The circular section being best suited to resist both internal and external pressures, tubes were naturally chosen as the means of separating the steam from the water, and small tubes admit of a very large amount of surface in a small space.

Copper, being highly ductile and one of the best conductors of heat, was at first chosen as the material from which to make the tubes. But it was soon found that the acids derived from the fatty matter from the cylinders dissolved some of the copper, and produced soluble salts of that metal, which were pumped into the boiler with the feed-water, and there caused great injury to the iron surfaces. This, for a time, gave the surface-condenser an ill repute, as it was found that the value of fuel saved by them was exceeded by that representing wear and tear of the boilers. This was eventually obviated by having the copper tubes coated with tin, and by discontinuing the use of tallow in the cylinders. But, notwithstanding this, the boilers in H.M. Navy continued to show signs of premature decay, such as was not customary with those receiving water from a jet-condenser. It was attributed to the highly corrosive power of redistilled water on the bare surface of the iron, together with the impossibility of keeping a protective scale on the surfaces when such water was used; later research has shown also that some gases which entered the boiler with the original water *chemically combined* with bases, which kept them comparatively innocuous, were freed, and came back *mechanically mixed* or in solution with the feed-water from the hot-well, capable of highly destructive action on iron surfaces, carbonic acid being a common and very corrosive one.

Copper tubes were, of course, expensive, and the tinning added to their cost very considerably, thereby rendering the first cost of a surface-condenser an appreciable addition to that of the engine. In 1860 brass tubes had long been used for boilers, and as this material was very ductile, and its galvanic action on iron feeble, it was tried as a substitute for copper in the manufacture of condenser tubes with at first mixed success, the want of complete success being partly due to want of care in manufacture, and partly to prejudice. The partial has since become a complete success, and all condenser-tubes are now made of brass, but somewhat richer in copper.

The Admiralty, and consequently all foreign governments, required brass tubes to be tinned when the condenser was of iron, and some of the large steamship companies also adopted this practice, but, as a rule, now the tubes are untinned. The tinning adds $1\frac{1}{4}$d. per pound extra to the cost of the tubes, and amounts to an increase of 16 per cent.; it is not necessary, but is an additional safeguard against the formation of copper salts on the one side, and to the corrosive action of sea-water on the other. Brass tubes, untinned, after twelve years' constant use, have been found, on being cleaned, to be nearly as good as when new; on the other hand, brass tubes have pitted badly, and in places been perforated in a few months from causes practically unknown.* It was conjectured that galvanic action had set up from iron filings, carried on to the tube surface by the steam or circulating water, causing a separation of the copper and zinc and the dissolving away of the latter. The investigations Prof. Bengough made for the Institute of Metals has thrown much light on the matter, and will eventually clear up what was somewhat of a mystery. As the condensers of warships are now always made of brass or steel, the tubes are untinned, but the Admiralty require the addition of 1 per cent. of tin to the mixture of copper and zinc, as this composition does not usually pit or corrode so readily as the ordinary brass mixtures have so often done.

The loss from blowing off the boilers to prevent dangerous incrustation when fed from the hot-well of a jet-condenser, amounted to as much as 25 per cent., and was seldom less in general practice than 12 per cent. This loss is almost wholly avoided by the use of a surface-condenser, and an additional saving of no mean importance is effected by cessation of necessity to scale and clean out the boilers as was the case when jet-condensers were used. *The net saving of fuel* by the use of a surface-condenser averages 15 per cent.; and in the hands of a careful engineer, the economy has been found to extend to even 20 per cent.

When sea-water is raised to a temperature of 280° F., which corresponds to a pressure of 50 lbs. absolute, or 35 lbs. above the atmosphere, what are called its *insoluble salts* are wholly precipitated, and these form a hard scale on the hot surfaces. The principal insoluble salt in sea-water (*v.* Chapter xxx.) is sulphate of lime; it is called insoluble, because it does not dissolve in water under ordinary circumstances, and consequently when deposited on the surface of the tubes, etc., will not easily redissolve and wash off again. The carbonate of lime, and the salts of soda and magnesia, are comparatively harmless, for although the former is precipitated, it is only in a soft muddy state, and mixed with the brine products of the latter can be blown off, and easily removed from the boiler when in port. It is for this reason that a surface-condenser is an imperative necessity for engines using steam above 35 lbs. pressure, or 50 lbs. absolute.

For the same reason it is advisable to fit a surface-condenser to steamers running in muddy fresh water, as otherwise the boiler soon fills with deposit, which, unless removed and the boiler thoroughly cleaned, will cause serious damage, and be a source of danger and expense, as well as a constant cause of priming.

It is seen, then, that by the use of a surface-condenser steam of higher

* *Vide* Reports of the Corrosion Committee of the Inst. of Metals.

pressure than 50 lbs. absolute may be employed with safety; a considerable saving of fuel and time is effected; and there is considerably less risk of burning and otherwise damaging the boiler. A better vacuum is also obtained in it, as a rule, than was possible with the jet-condenser. On the other hand, a surface-condenser is heavier, more costly, and occupies more space than the jet-condenser; a second pump for the cooling water is necessary, and although the air-pump need not be so large as for a jet-condenser, it was often made so in case the jet was used or the tubes leaked. It was urged that the wear and tear and the store account are increased somewhat when there is a surface-condenser, and more care and responsibility are laid on the engineers; but these are mere trifles compared with the benefits, and are only mentioned now as examples of the arguments used by the opponents of Mr. Hall's system in the bitter controversy that raged in the forties of last century, and prevented its general adoption. For sea-going vessels the surface-condenser is indispensable, and now always fitted, notwithstanding its weight and other drawbacks.

Condenser efficiency was never, as a matter of fact, seriously considered until quite recent times, for the old school of engineers had other distractions, and were quite content with what is now considered to be only a moderate or even a low vacuum, especially were they complacent if with it was a hot-well temperature of 120° F., without enquiring very closely as to how it was obtained. Moreover, so long as it was accepted as an axiom that any higher vacuum could be of no service to the marine engine, there was no incentive to enquire into, much less to attempt better things. The advent of the turbine as a marine motor, however, together with the demand for the highest speed with some ships at any reasonable cost, has changed all this, and turned the apathy of the past into the activity of the present with regard to the condenser. Moreover, engineers have been impressed with the apprehension that while little remained to be gained at the upper part of the expansion diagram, at the lower the field was large and fallow. Thoughtful minds have turned, therefore, to it, and made a careful study of the theory and practice involved in the working of a surface condenser. To Mr. James Weir, Mr. Morison, Prof. Weighton, and others, we are much indebted for the better knowledge we possess, and although these gentlemen do not always agree they are all equally frank and communicative of what they discover when experimenting, so that to maintain 29 inches vacuum in a modern condenser in the temperate zone is easy. To the turbine such a small back pressure is of the utmost value, whatever it may be to a reciprocator, and its efficiency with it very high. But it must not be taken now for granted that no advantage can accrue to a reciprocating engine by the reduction in back pressure. No doubt, owing to restriction in the valve and passage sections existing in most of these engines, the high volume steam has to flow at such excessive speeds that the difference in pressure between the cylinder and condenser is necessarily greater than required between a turbine and its condenser, or even what obtains when the back pressure is greater. At 29 inches of vacuum, or a half-pound pressure, a pound of steam has a volume of 640 cubic feet, while at 26 inches it is only 172; for the same weight of steam the velocity of flow to the condenser will, therefore, have to be nearly four times at the high vacuum if the pressure difference of it is to be the same as at 26 inches. But it is quite unlikely that, with a decrease in the pressure in the condenser, there will be none in the cylinder, whatever its ports and passages be. That there is some decrease is certain, and such decrease will enable a greater power to be developed; and that the decrease may be considerable may be seen on examining the diagram shown in fig. 118*b*. Each pound of back pressure means $2\frac{1}{2}$ to 3 per cent. difference in the power developed by a triple- or quadruple-compound engine, and such an increment as this is quite possible in a properly designed engine by

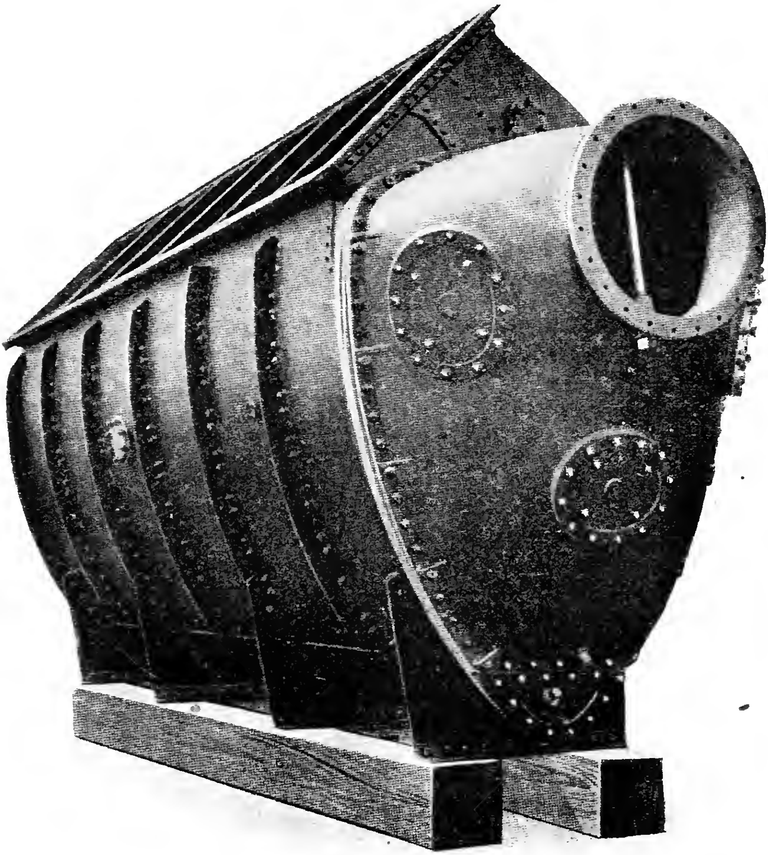


Fig. 115.—"Weir-Uniflux" Condenser for a Turbine of 11,000 S.H.P.

merely increasing the vacuum from 26 to 29 inches. Moreover, if marine reciprocating engines are designed with a multiplicity of cylinders that each may be kept of small size, as they are on the lines of the oil engine, so that instead of one or, at most, two L.P. cylinders per engine there are three or four, then the port and passage ratios may be so much larger that high vacuum would be of advantage to such engines.

The effect on economy of consumption is another matter, and has to be considered from quite a different standpoint. More energy will be required by the air-pumps to maintain such higher vacuum; and as the temperature due to 29 inches is 80°, against the 126·8° of 26 inches, the feed water must leave the condenser so much cooler; but whether the reheating will be more costly than the gain is rather a matter of experience and trial than of calculation; for so great a difference probably it is so; but as 28 inches can be much more cheaply obtained than 29 inches, and the temperature difference is then only 25° against 46·8° as above, the loss, if really anything, cannot be great in these days of feed heaters.

Mr. Weir attaches no importance to the question of water deposit on the tubes in a surface-condenser, and thinks there is nothing to be gained, while there is something lost, by fitting, collecting, and training plates among the tubes, whereby it is conducted away as soon as formed without further

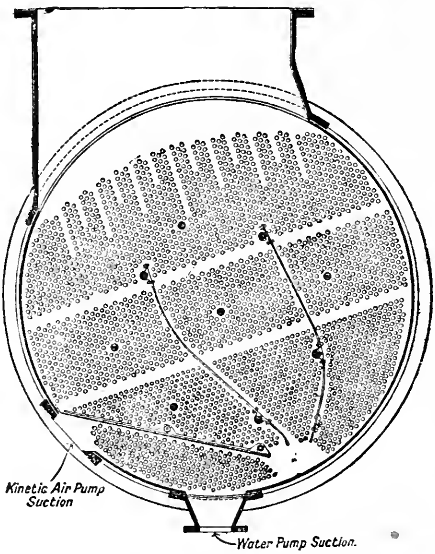


Fig. 116.—Cylindrical Condenser on Morison's System

contact; whereas Mr. Morison advocates and fits them to all his condensers, and claims to get thereby a distinct benefit. The former authority is, however, an equally strong advocate for having warm hot-well water, and he should, therefore, not object to any reasonable natural means of attaining that very desirable end. With that end in view, the exposure of the water when once formed to a further cooling action cannot be helpful; on the contrary, the already condensed water must lose some of its own heat, and add to that of the cooling water, thereby reducing its efficiency. Further, by means of such plates as Mr. Morison and others fit, the steam and air are dispersed over the whole of the cooling surface without scattering the water when condensed. But, on examining the design of these gentlemen's condensers, one is struck by the necessity for such arrangements in the one and its absence of it in the other. Mr. Weir's condenser is of triangular cross-section or heart-shaped, as an approximation to it (fig. 115). Steam enters at the butt end through such an enormous orifice as to enable it to spread at once over the whole breadth and length (which is generally limited) of the top of the condenser, and gravitate to the contracted bottom as it cools and shrinks in volume. In this case the water, as it forms, can, and does, drop on to the inclined sides, on which it courses to the bottom, which is not very far distant. Here there is little chance of any of the water dripping on to the colder lower rows of tubes from the majority of the upper ones. But the shape of the condenser is not economical of space or suitable to resist unaided external pressure. On the other hand, the cylindrical condenser of Mr. Morison (fig. 116) only requires inexpensive guide plates, as does also that of Mr. Morcom, which shall cause a perfect circulation of the steam among the tubes and the leading away of the water when formed is accomplished by the same means surely with some advantage. In the case of the compact rectangular section condenser forming part of the engine, and economic of room and cost, as in common mercantile practice, the Contraflo system (fig. 117), with the similar drainage scheme, is also advantageous —in other words, circumstances alter cases. Fig. 117*a* is a diagrammatic form of a Contraflo, and shows how the actual one, as in Fig. 117, is virtually the same as the wedge-shaped or triangular section one.

For efficiency, a complete and rapid circulation of the steam over the whole of the cooling surface is essential, as is also the concentrating of the air and water into a natural flow towards the passage to the air-pump; the simplest means of effecting these two things are probably the best.

Equally important is the question of cooling-water distribution. Its quantity depends on the difference in temperature between it as it enters and that of the condenser where it leaves it, the weight of steam to be condensed, and the rapidity of flow. In practice condenser tubes are usually $\frac{3}{4}$ inch in external diameter in the mercantile marine, and $\frac{5}{8}$ inch on naval ships. Now, a tube $\frac{3}{4}$ inch diameter, 18 gauge thick, and 10 feet long, has a surface of 1·95 square feet, and inasmuch as 30 lbs. of steam per square foot per hour may be condensed on it when clean, it will be necessary for a vacuum of 28 inches that about 1,200 lbs. of cooling water should pass through it per hour in winter time in the temperate zone, and 3,600 lbs. in the tropics; the flow then will be at the rate of 400 feet per minute in the latter case, which is somewhat excessive, and would require considerable power to obtain, as the friction head would be about $3\frac{1}{2}$ feet per tube length. On the other hand, in the temperate zone in winter the flow would be too moderate.

This means that if a good vacuum is to be maintained in the tropics the tubes must not be very long with so small a diameter, while, on the other hand, if a ship is never to be where the cooling water is above 60°, the tubes may be long with advantage. As, however, the rate of flow will vary as the square of the diameter of tube, while its surface is directly as the diameter, a small increase in diameter will admit of considerable increase in length.

In practice, the cooling water may have sufficient heat imparted to it to raise it to a temperature very little below that of the condenser top before leaving it. As this warming up, however, is usually done in stages—that is, the water passes and repasses through the tubes on its journey from the entering in to the leaving them—three stages is the most common, so that in a general way the increment added to the temperature of the circulating

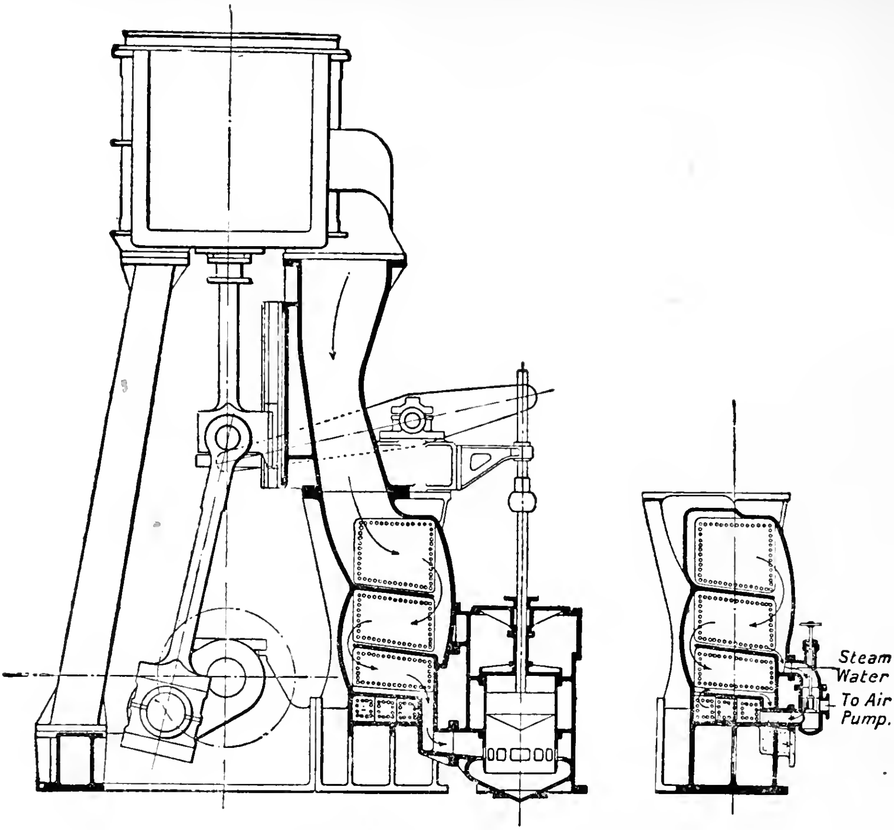


Fig. 117.—Ordinary Marine Engine with Contraflo Condenser.

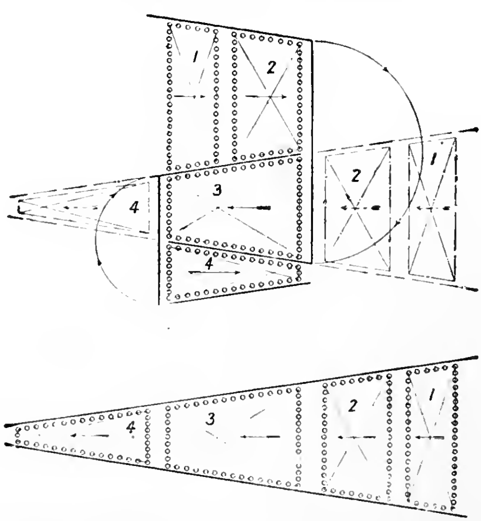


Fig. 117*a*.—Diagram showing the Flow of Steam in the Contraflo System.

water at each pass should be only one-third of the total increase. To do this the number of tubes in the lower or first group should be much less than that in the second, and less there than in the third, as the efficiency owing to difference in temperature between it and the condenser is decreasing at each stage. This circumstance accentuates the reason for shorter tubes or of larger diameter when the water is warm and a high vacuum desired.

Surface Condenser Efficiency has, however, become a matter of first-rate importance ever since the introduction of the turbine on shipboard, for it was there discovered how seriously the efficiency of that instrument was augmented when working with the very low back pressures obtainable there at comparatively cheap rates, owing to the unlimited supply of cooling water obtainable with little sacrifice of power due to the small resistance "head." Mr. Parsons, and those interested in the success of the turbine, turned their attention to improving the means whereby high vacuum could be obtained and maintained; Mr. Morison, ably assisted by Prof. Weighton, of Durham, followed on with a series of most interesting and instructive experiments, and their investigations have been the means of throwing quite a flood of light on the subject, and clearly demonstrating exactly what takes place in a condenser and its pumps.

The effect of Air mixed with Steam or water vapour had been noted by Prof. Osborne. Reynolds, and others years before, and it was well known that the presence of air was always a cause of retardation of condensation, and the reduction of amount of water deposited on a cold surface in a fixed time. If steam alone entered a surface-condenser, it would be wholly transformed into water, and a vacuum corresponding to the temperature would be maintained in it. The amount of steam condensed per square foot of surface would be very high, so long as the tube remained clean inside and outside and the cooling water supply plentiful with the flow through the tubes rapid. The only pump necessary under these circumstances would be one to withdraw the condensed water as it falls to the bottom. But, as a matter of fact, it is impossible in practice to work with a circuit so completely closed that no air gets into the system when once the water put into the boilers has been deprived of the air it originally held in solution, for every condenser of a marine engine contains more or less air always, and, therefore, an air-pump is necessary to it, in order to maintain any sort of steady vacuum; and if the vacuum is to be high, the air-pump must be efficient as well as sufficiently large; for, however large it may be, if it is not efficient no high vacuum at all is possible.

Air Leaks to the Condenser may have originated at the glands of the main and auxiliary engines when the pressure inside them is less than that of the atmosphere. Much air comes from the feed-water, which, if taken from storage tanks, will contain from $2\frac{1}{2}$ to 4 per cent. by volume of air. Even the water of the hot-well is charged with a considerable amount, and this Mr. Weir endeavours to eliminate in his feed-heaters before it can enter the boiler. Auxiliary feed-water is subject to the same action in the feed-heaters. Leakage may, and does, arise at times from faulty or damaged jointings of the condenser and connections; these leaks, however, should be found out and stopped; for this purpose a periodical examination is made by some careful engineers in charge of turbines; those with reciprocators should do the same. The leakage from auxiliary machinery, however, is often a most serious cause of loss, and is said to amount to as much as the cost of the whole power developed in them. No doubt the Admiralty are wise to have separate condensers for the auxiliary machinery, and in large merchant steamships the same practice might be followed with considerable advantage; even if such condensers were simplified so that the water circulation was "natural," and the condensed water allowed to drain into a tank by gravity, it is better to have them rather than to run the risk of spoiling the main condenser efficiency by admitting aërated auxiliary exhaust to it.

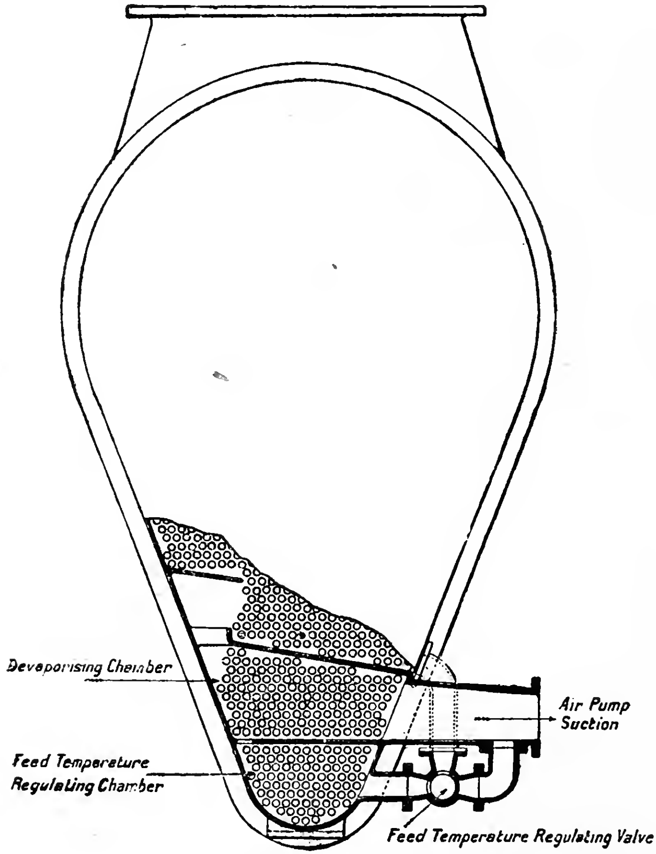


Fig. 118.—Contraflo Condenser with Feed Temperature Regulator.

Air is heavier than Water Vapour, so that, if the exhaust steam contains but a very small quantity, it will accumulate at the bottom of the condenser, and unless removed it practically blankets the cooling surface there, and virtually reduces the size of the condenser. Even when the air-pump draws it away, if only at the same rate as it flows in, there may be a portion of the surface constantly surrounded by air and effecting no condensation. It is necessary, therefore, that the air-pump shall be large enough to keep the inside of the condenser free from air lodgment.

It follows, then, that, in addition to the large efficient pump, means shall be provided that the flow to it will be as direct as possible; and at the

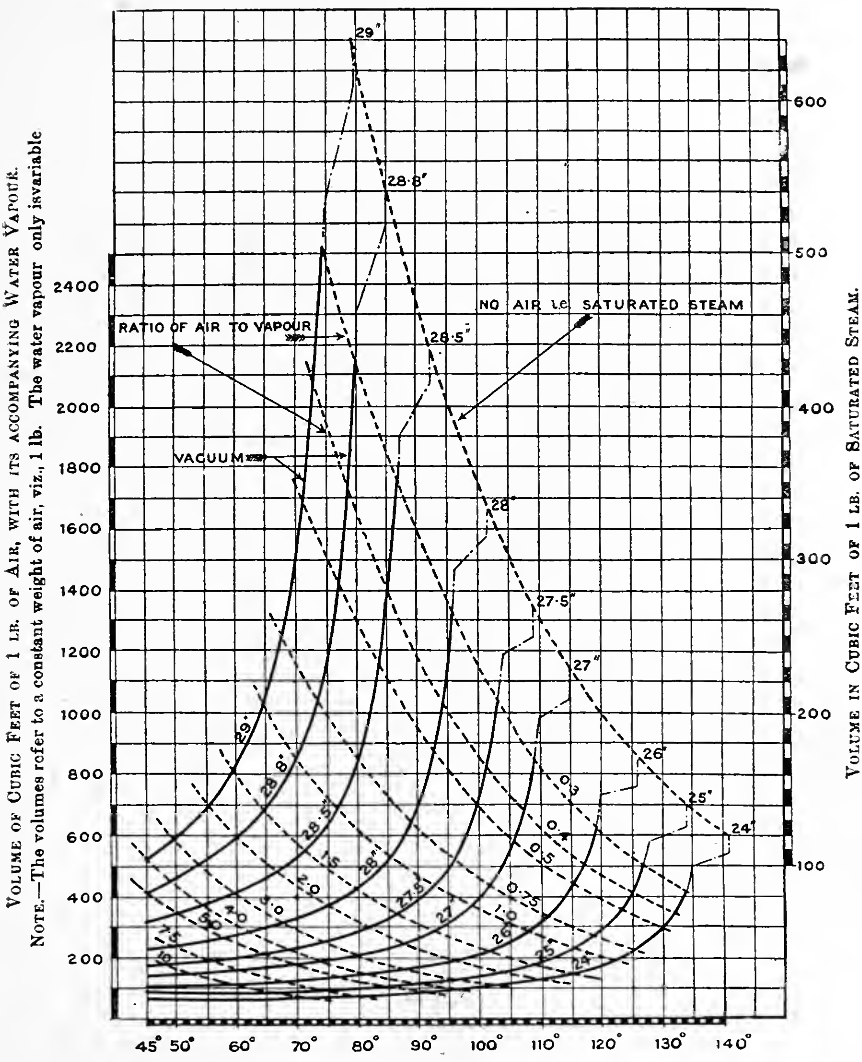
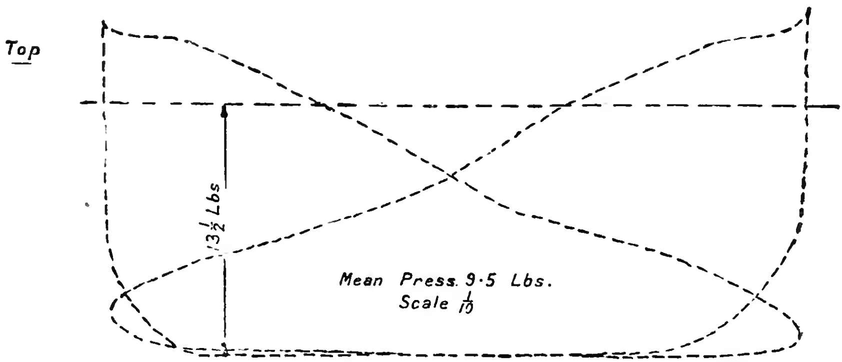


Fig. 118a.—Morison's Diagram. Air Saturated with Water Vapour.

Curves showing relation between vacuum, temperature, volume, and ratio of air to vapour.

same time that every part of the cold surface should be active; hence there must be no pockets or eddies anywhere, and if the design of condenser does not ensure this constant and general flow, the vapour must be directed by guide plates and baffles. Mr. Morison has given great attention to this, and all else that pertains to the surface-condenser, and been good enough to impart his knowledge in the papers he has read at the meetings of the Institution of Naval Architects, and of the N.E. Coast Institution of Engineers and Shipbuilders, the transactions of which may be studied with advantage.

The Flow of Vapour and Water in its passage through the condenser from the exhaust to the air-pump suction must be at all times positive, but separate. When water is once formed, it should not be kept in contact with the cooling surface, or allowed even to touch it again, but at once take its passage to the drain. The vapour should be made to pass or repass over the whole cooling surface, so that no part of it is inactive. In modern condensers the continuous contact is done by repassing largely in order to economise the space taken by the condenser, which makes it virtually the same as a long one of wedge shape, transverse section, like Mr. Weir's (fig. 117), having the butt end next the cylinder, and the drain to the air-pump at the thin end. In this way the air is concentrated before entering the pump, and is moreover cooled as much as is possible, so that the air-pump



Steam, ...	176 lbs. per square inch.	High-pressure engine, ..	630	I.H.P.
Vacuum, ..	28¼ inches.	Intermediate-pressure engine,	697	,,
Revolutions, ..	66 per minute.	Low-pressure engine, ...	708	,,
		Total, ..	2,035	,,

Fig. 118*b*.—L P. Indicator Diagram showing High Vacuum. S.S. "Nigaristan."
(Richardson and Westgarth.)

can be kept cool and fit to produce in its chamber a very high degree of vacuum. For this purpose the lowest part of the condenser is often arranged so that the lowest rows of tubes through which the coldest water passes are "drowned," so that the water which surrounds them is cooled down below the temperature of the bulk of the condensed water.

Fig. 118*a*, Morison's diagram showing the relation between vacuum, temperature, volume, and ratio of air to vapour, is a most interesting and instructive one, and may be referred to with advantage when considering condenser problems.

Cooling Surface.—Mr. Weir has shown that, with efficient air-pumps, as much as 16 lbs. of water can be condensed on a square foot of cooling surface per hour in a vacuum of 26·75 inches when the sea-water is as hot as 80° F., and the hot-well temperature as high as 106°. With sea-water at 54° F., and the vacuum 28 inches, as much as 28·6 lbs. were condensed per square foot per hour. He also demonstrates how as much as 35 lbs. can be condensed with the vacuum at 27·3 inches, with the cooling water at the same temperature, but with the hot-well temperature reduced to 101°, or nearly that in the condenser.

It would seem, then, from these facts that 1 square foot of cooling surface per I.H.P. is sufficient for any ship, so long as the condenser is fairly clean; and, further, it is evident that ½ square foot per I.H.P. is ample allowance for such ships as work in temperate zones with cooling water under 60°. Moreover, as most ships on service run with less power than developed on trial, the allowance for trial conditions practically provides a margin for loss of efficiency of surface afterwards. Experience with turbine-driven ships, having a vacuum in their condensers of 28·5 to 29·0 inches, has established the fact that with cooling water at 50° F. and the hot-well kept at 80° to 85° F., as much as 25 lbs. of steam can be condensed per square foot of surface per hour.

The Allowances of Cooling Surface may be computed in the following way:—Ships trading to all parts of the world, and, therefore, sometimes in the tropics, should have 1 square foot of cooling surface for each 16 lbs. of steam condensed. Turbine steamers working under the same conditions should, to maintain high efficiency, have 1 square foot for every 12 lbs.

Ships limited to service in temperate zones, or whose service in the tropics is short, or when there high efficiency is not of consequence, 25 lbs. for reciprocators and 20 lbs. for turbines is not too large an allowance. The temperature of the condenser with a vacuum of 29 inches is 80° F.; it is obvious, therefore, that, with sea-water at this or a somewhat higher temperature, such a vacuum is impossible, whatever be the surface, and even with 28 inches the amount of cooling water required would be so great as to be almost prohibitive in commercial practice.*

The temperature of the condenser is somewhat higher near the entering in of the steam than at the bottom when the engine is working at full power, but at reduced power the difference is very trifling; at full speed, therefore, the circulating water may be nearer the temperature at bottom of condenser than at low powers. The water should, therefore, for this and other reasons, always flow in the opposite course to that of the steam—that is, the coldest water should be where the steam is coldest, and where the steam is entering the condenser, and, therefore, at its hottest the heated cooling water may still take up more heat from it, leaving it with heat still in it for abstraction below.

The Cooling Surface per Horse-power may be as follows:—

Triple-compound express steamers,	0·80 sq. ft. home waters,	1·25 sq. ft. tropics.
,, ,, economic	,,	0·70 ,, ,,	1·06 ,, ,,
Quadruple ,, ,,	,,	0·65 ,, ,,	0·95 ,, ,,
Turbine-driven ordinary	,,	0·65 ,, ,,	1·10 ,, ,,
,, express	,,	0·80 ,, ,,	1·25 ,, ,,

For ordinary cargo steamers going to all parts of the world 1 square foot of surface per trial trip I.H.P. is sufficient, as the temperature of sea-water does not exceed 85° F., and is not often over 80°. This allowance is sufficient to permit of good vacuum being maintained on service conditions

* At 80° F. the cooling water must be 63 times the weight of steam; and at 85° no less than 153.

with the surface somewhat foul. Destroyers with reciprocating engines on trial in summer time have condensed 28 lbs. of steam per square foot, and the allowance per I.H.P. was only $\frac{1}{2}$ square foot, and the vacuum 25 to 26 inches at full power.

Professor Weighton, with a Contraflo condenser, got 33 lbs. condensed per square foot from a triple-compound engine; and even 40 lbs. have been got by other experimenters.

Cooling Surface.—Professor Rankine suggested the following as a means of ascertaining the amount of cooling surface:—Let t denote the temperature of a film of liquid at one side of a metal plate, S the extent of cooling surface; let heat be communicated to the liquid at a temperature t by some such process as the condensation of steam, and let that be abstracted by the flow of a current of air, water, or other fluid, in contact with the metal plate; the weight of fluid which flows past per second being W, its specific heat C, its initial temperature T_1, being lower than t, but higher than T_2. Then in all the equations $t - T_1$ is to be substituted for $T_1 - t$, and $t - T_2$ for $T_2 - t$ in the equation $\frac{S}{c\,W} = a\left\{\frac{1}{T_2 - t} - \frac{1}{T_1 - t}\right\}$; but he also added that there are not sufficient data to obtain the value of the constants.

Another Basis for Calculating Cooling Surface is obtained by assuming that the maximum mean flow of cooling water permissible is 400 feet per minute, and the units of heat to be abstracted being that latent at the different pressures, and the cooling water in the tropics 80° F., and in home waters 60°. The diameter of the condenser tubes is taken as $\frac{5}{8}$ inch for naval practice, $\frac{3}{4}$ inch in general mercantile, and 1 inch in exceptional cases.

The weight of water passing per minute through a $\frac{5}{8}$-inch tube at this rate will be 44 lbs., through a $\frac{3}{4}$-inch 60 lbs., and through a tube 1 inch in diameter 107 lbs. The temperature of the water at discharge from the condenser will be taken at about 2 per cent. below that of the hot-well water, or that of the condenser, so that for a vacuum of 28 inches it will be 0·98° × 90·4° = 88·6°. Under these circumstances, in the tropics the heat abstracted will be 88·6 — 80, or 8·6°. The latent heat with this vacuum will be 1,051°; hence the weight of cooling water per lb. of steam $= \frac{1,051}{8 \cdot 6} = 122$ lbs.; a tube $\frac{5}{8}$ inch in external diameter will condense, therefore, in an hour $60 \times \frac{44}{122}$ or 21·6 lbs.

The greatest length of tube through which the water flows should not exceed 400 diameters, and, therefore, if the water passes three times, as is common practice, the $\frac{5}{8}$-inch tube should not be in the aggregate more than $(\frac{5}{8} \times \frac{400}{12})$, or 21 feet, and each tube is consequently only 7 feet long, or if only twice through 10·5 feet.

Now, 21 feet of $\frac{5}{8}$-inch tube has a surface of 3·437 square feet. It follows, then, that the cooling surface in a condenser having $\frac{5}{8}$-inch diameter tubes for service in the tropics when 28 inches vacuum is required must be at the rate of 1 square foot for each, 21·6 ÷ 3·437, or 6·3 lbs. of steam to be condensed.

Taking the consumption of steam in a turbine steamer at 12·5 lbs. per S.H.P., then—

Cooling surface of tubes $\frac{5}{8}$ inch in diameter per S.H.P. for tropics 28 inches vacuum is 2 square feet.

If sea-water is 60°, there will be then the following, viz. :—
Heat abstracted 88·6 — 60, or 28·6°.
Weight of cooling water = 1,051 ÷ 28·6, or 36·7 lbs. per lb. of steam.
Water condensed per hour = 60 × 44 ÷ 36·7, or 72 lbs.
Rate of condensation per square foot per hour = 72 ÷ 3·437, or 21 lbs.
For home service, therefore, 12·5 ÷ 21, or 0·596 square foot per S.H.P., is sufficient for $\frac{5}{8}$-inch diameter tubes.

For the Mercantile Marine with $\frac{3}{4}$-inch tubes and a 28 inches vacuum, the following will hold good, viz. :—

Heat abstracted in the tropics, as before, 8·6 units.
Water passed per hour, 60 × 60, or 3,600 lbs.
Weight of cooling water per lb. of steam, 1,051 ÷ 8·6, or 122 lbs.
Weight of steam condensed per hour = 3,600 ÷ 122, or 29·5 lbs.
Maximum length of tubes, $\frac{3}{4} \times \frac{400}{12}$, or 25 feet.
Each tube being 8·33 feet long if three times through, or 12·5 if twice.
The surface of 25 feet of $\frac{3}{4}$-inch diameter tube is 4·91 square feet.
Quantity of water condensed per square foot is then 29·5 ÷ 4·91, or 6 lbs.

If, however, the condenser were made the same length as that with $\frac{5}{8}$-inch tubes, the surface would be as before 3·437 square feet, and the quantity of water condensed per square foot 29·5 ÷ 3·437, or 8·6 lbs. under these circumstances.

The cooling surface per S.H.P. = 12·5 ÷ 8·6, or 1·45 square feet.

If the condenser tubes are made 1 inch in external diameter, the water passed per hour is 60 × 107, or 6,420 lbs.; weight of cooling water as before, 122 lbs.

Steam condensed per hour, 6,420 ÷ 122, or 52·6.
Maximum length of tube = $1 \times \frac{400}{12}$, or 33·3 feet.
So that in this case each tube may be 11·0 feet long.
The surface of 33 feet of 1-inch tube is 8·64 feet.

Quantity of steam condensed per square foot = $\frac{52 \cdot 6}{8 \cdot 64}$, or 6·1 lbs.

If the cooling water has a temperature of 60°, the number of heat units abstracted by each pound will be as before 88·6 — 60, or 28·6°.

The weight of water for each pound of steam, $\frac{1{,}051}{28 \cdot 6}$, or 36·7 lbs.

Steam condensed = 6,420 ÷ 36·7, or 175 lbs.
Steam condensed per square foot = 175 ÷ 8·64, or 20·3 lbs.

From the above it will be seen that if the maximum combined length of 400 diameters of tube is followed as the rule for condensers, that a vacuum of 28 inches should be maintained with a rate of condensation of about 6 lbs. per square foot of surface when in the tropics, and 20 lbs. in the temperate zone. For the steam consumption per horse-power this would indicate that the cooling surface for tropical work must be at least three times that for cool climates when high vacua are required and necessary as with turbines. It has been, however, pointed out that under tropical conditions much more than 6 lbs. of steam per square foot of surface can be condensed. It follows, then, that the length of tube must be reduced, and the combined length be inversely as that quantity is to 6.

That is, if Q be the quantity in pounds, then—

$$\text{Combined length} = 400 \times \frac{6}{Q} \times \frac{d}{12}.$$

d being diameter of tube in inches.

If, however, a vacuum of 26 inches is sufficient, as is the case with reciprocating engines in the tropics, then the temperature of condenser will be 120°, and the latent heat 1,030°.

The number of units abstracted per lb. of cooling water will be 120 — 80, or 40.

Weight of water for each pound of steam, 1,030 ÷ 40, or 25·75.

Taking the tubes $\frac{3}{4}$ inch in diameter—

The steam condensed = 3,600 ÷ 25·75, or 140 lbs.

$$\text{The steam condensed per square foot surface} = \frac{140}{4{\cdot}91} = 28{\cdot}5.$$

It is evident from the rates observed by Mr. Weir and others that the high vacua are obtained in the ordinary ship's condenser; it follows, then, that either the tubes were short or the rate of flow through them much higher than 400 feet per minute or both. The difference in temperature has been taken here at only 8·6° F. for the tropics, which is, of course, very low. But if long tubes are fitted to a condenser the rate of flow must be very high to maintain high vacua, and the allowance of surface more than shown above. If, therefore, 28 inches is to be maintained with 1·25 square feet of cooling surface per S.H.P., the flow of water through the condenser, whose tubes are 400 diameters in length, at a rate per minute $= \frac{400 \times 2}{1{\cdot}25}$, or 640 feet, which is high for any tube, but especially so for those only $\frac{5}{8}$ inch in diameter. The "head" to overcome the mere resistance of 21 feet of such tube will be at a velocity of 640 feet per minute, probably as much as 45 feet, while at 400 feet it will be no more than 16·4 feet.

With tubes $\frac{3}{4}$ inch in external diameter and 25 feet aggregate length the "head" to overcome resistance will be 15·7 with a velocity of 400 feet, and 35·7 feet if the velocity is raised to 600, while with tubes of 1 inch external diameter, and an aggregate length of 33 feet, the resistance at 600 feet per minute flow will be only 14 feet "head."

TABLE XL.—EFFECT OF VACUUM ON STEAM CONSUMPTION IN LBS. PER I.H.P. IN A TURBINE AND QUADRUPLE-EXPANSION ENGINE (FROM PROF. WEIGHTON'S OBSERVATIONS), AND IN A TRIPLE-EXPANSION ENGINE (200 I.H.P.) AND 1,000 K.W. TURBINE. (FROM SIR C. PARSONS' OBSERVATIONS.)

Vacuum, . . Ins.	20	22	24	26	27	28	29
Turbine, Lbs. per H.P.	19·2	18·1	16·9	15·6	14·8	13·9	13·0
Quadruple, ,,	16·7	16·1	15·5	15·0	14·7	14·5	14·3
Triple-comp. eng. ,,	14·8	14·35	14·05	13·90	13·8	13·77	13·76
Turbine ,,	19·3	18·1	16·9	15·6	14·9	14·0	13·0

TABLE XLI.—TEMPERATURE, LATENT HEAT, AND VOLUME OF STEAM OF VERY LOW PRESSURE.

Pressure. Lbs. Absolute	Pressure. Vacuum.	Temperature F°.	Latent Heat F°.	Volume of 1 Lb. of Steam. Cub. Ft.	Pressure. Lbs. Absolute	Pressure. Vacuum.	Temperature F°.	Latent Heat F°.	Volume of 1 Lb. of Steam. Cub. Ft.
0·3,	29·4	67·5	1,070	1,067	1·7,	26·6	120·3	1,029	200
0·4,	29·2	74·0	1,063	800	1·8,	26·4	122·4	1,028	190
0·5,	29·0	80·0	1,058	640	1·9,	26·2	124·6	1,026	181
0·6,	28·8	85·5	1,054	535	2·0,	26·0	126·7	1,025	172
0·7,	28·6	90·4	1,051	461	2·1,	25·8	128·6	1,024	165
0·8,	28·4	94·5	1,048	410	2·2,	25·6	130·4	1,022	158
0·9,	28·2	98·5	1,045	367	2·3,	25·4	132·2	1,021	152
1·0,	28·0	102·0	1,042	333	2·4,	25·2	134·0	1,020	146
1·1,	27·8	105·0	1,040	306	2·5,	25·0	135·6	1,019	140
1·2,	27·6	108·0	1,038	282	2·6,	24·8	136·9	1,018	135
1·3,	27·4	111·0	1,036	260	2·7,	24·6	138·2	1,017	130
1·4,	27·2	113·7	1,034	240	2·8,	24·4	139·6	1,016	125
1·5,	27·0	116·0	1,033	225	2·9,	24·2	141·0	1,015	121
1·6,	26·8	118·2	1,031	212	3·0,	24·0	142·2	1,014	118

Condenser Tubes.—They are, as a rule, made of brass, solid drawn, and tested both by hydraulic pressure and steam; the latter test is a very useful one, as faults which escape detection under water pressure are often found out by steam; these faults are due generally to minute particles of flux or slag in the original ingot, and sometimes the faults are in the form of cracks done in the process of drawing. It is by no means an uncommon thing to find a few tubes in a new condenser leaking through minute holes of various shapes; these holes soon become enlarged if the tube is not at once stopped or withdrawn. These faults are not confined to the tubes of a few makers, but may be found in those of all makers at some time or other. Tinning is, as a rule, a preventive, as the defective places are in the process covered or filled with that metal, but it is seldom resorted to now. The Admiralty method of adding a small amount of tin to the alloy of copper and zinc has proved a good preventive of corrosion, and Mr. Philip, Admiralty Chemist, has shown by statistics its efficacy in a large number of condensers in H.M. Navy.

Condenser tubes were usually made of a composition of 68 per cent. of best selected copper, and 32 per cent. of best Silesian spelter.* The Admiralty, however, always specify the tubes to be made of 70 per cent. of best selected copper, and to have 1 per cent. of tin in the composition, and test them to a pressure of 300 lbs. per square inch. To prove that the tubes are of the 70/30 alloy, a few pounds of them are melted in a closed crucible, and sufficient spelter added to bring the mixture to contain 62 per cent. of copper. The metal is then cast into an ingot, which when cold is rolled into a sheet, strips are cut from it and tested, and if satisfactory should have an ultimate tensile strength of 24 tons per square inch. In the mercantile marine the tubes are, as a rule, $\frac{3}{4}$ inch diameter externally, and 18 L.S.G. thick (0·049 inch); and 16 L.S.G. (0·065), under some exceptional circumstances. In H.M. Navy, the tubes used to be 18 to 19 L.S.G. thick, tinned on both sides;

* The tubes of the mercantile marine are usually 70/30 mixture without additions, but latterly 2 per cent. of lead has been tried with success.

but now the Admiralty do not require the tubes to be tinned. On account of the economy of space and weight that is effected with small tubes all Naval engines are now fitted with condensers having tubes $\frac{5}{8}$ inch diameter. The smaller the tubes, the larger is the surface which can be got in a certain space. Since larger tubes are of necessity somewhat thicker than the smaller ones, a square foot of surface costs more when they are adopted, and is not so efficient. Patent tubes made from sheet brass 22 B.W.G. thick, and joined at the seams like a tinsmith's joint and soft soldered, have been tried. The advantage claimed for them is the uniformity, whereby as little as 22 B.W.G. is sufficient thickness, while it would not be safe to use drawn tubes so thin.

The length of the tube depends on the arrangement of the condenser, but when they are not held tightly in the plates, but only packed, their unsupported length should not exceed 100 diameters; when held with tight-fitting ferrules it may be 120 diameters.

Tube-plates are now always made of brass, either cast or rolled into plates of suitable size; the latter is preferable, as the rolled brass is very tough and close grained, and as strong as wrought iron. Formerly it was no uncommon thing to make the tube-plates of cast iron from $1\frac{3}{4}$ to $2\frac{1}{2}$ inches thick, and while some were converted into a substance resembling plumbago after two or three years' work, others have been found sound and good after twelve years' continuous work.

Rolled brass tube-plates should be from 1·3 to 1·5 times the diameter of tubes in thickness, depending on the method of packing. When the packings go completely through the plates the latter, but when only partly through, the former is sufficient. Hence, for $\frac{3}{4}$-inch tubes the plates are usually $\frac{3}{4}$ to 1 inch thick with glands and tape-packings, and 1 to $1\frac{1}{4}$ inches thick with wooden ferrules. In the Navy the tube-plates are generally 1 to $\frac{7}{8}$ inch thick, the tubes being $\frac{5}{8}$ inch diameter and 19 L.S.G. thick, but in the "Destroyers" the plates are only $\frac{3}{4}$ inch thick; in their case, however, the plates are of small diameter and stayed in the middle; and it may be added that leakage of tubes was no uncommon occurrence in such ships, so that this is as thin as they can safely be employed.

The tube-plates should be secured to their seatings by brass studs and nuts, or brass screw-bolts; in fact, there must be no wrought iron of any kind on the sea-water side of a condenser. When the tube-plates are of large area it is advisable to stay them by brass rods, to prevent them from bulging or collapsing.

Tube Packings.—All attempts to drift or expand the tubes tightly into holes in a brass plate fail, owing to the softness of both plates and tubes; and if it could be done it would be found impossible to draw the tubes for examination and cleaning without damage. Fig. 119 shows a very simple plan, and one that proved effective under all circumstances, and essential with cast-iron plates. The ferrule is made of soft wood, such as pine or lime tree, very dry and well seasoned; they were made nearly an eighth of an inch larger in diameter than the hole into which they had to fit, and a good fit on the tube. Before fitting them into place they were squeezed through a die in a press until they could be easily driven into their holes; soon after being fitted into place they absorb moisture and expand circumferentially at each end, and become exceedingly tight on the tube and in the hole. After twelve years' service they were found quite sound. It is urged against them that they are apt to shrink and drop out when the condenser is not in use, but this is not the case, as the swelled projecting ends form collars to prevent this, and they do not shrink so much as is generally supposed, unless by unusual heat. This is one of the cheapest forms of tube-packing, and although not used now, was often employed in the mercantile marine of this and other countries.

The plan adopted in H.M. Navy, and generally in the mercantile marine, is that shown in fig. 120. Each tube end passes through a stuffing-box fitted

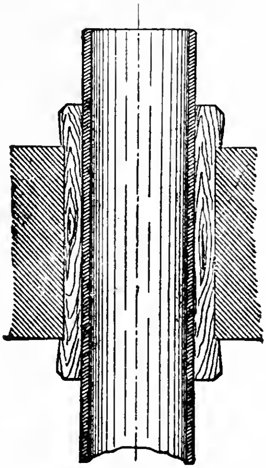


Fig. 119.

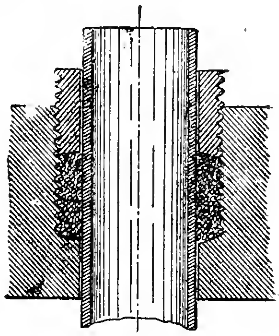


Fig. 120.

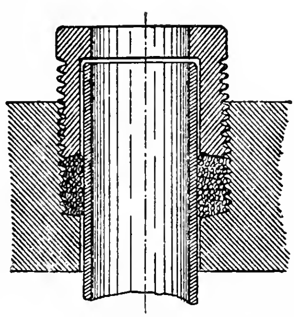


Fig. 121.

Figs. 119-121.—Condenser Tube Packings and Ferrules.

with a screwed gland, and kept tight by a tape washer, or some soft cord as packing. This method is somewhat expensive, but it admits of the water being on either side of the tubes; the packing is not affected by heat, and the condenser may remain unused for a very long time, and be quite tight at the end of it; for these reasons it was chosen by the Admiralty. It is likewise the plan used by Hall in his early surface-condensers, and on the whole the most satisfactory one. Modern methods of drilling and tapping the holes, making the ferrules, etc., have very much reduced the cost, and so practically removed the only objection. Fig. 121 shows a modification of gland specially to suit vertical tubes; the gland has an inside rim, which prevents the tube from slipping. It is now always used in condensers with horizontal tubes, and is especially necessary when they are long to prevent creeping.

Steam Side and Water Side of Tubes.—This was somewhat of a vexed question, about which there is much to be said on both sides. The naval practice was formerly to circulate the water *outside* the tubes, so that the condenser shell may be kept cool and prevented from making the engine-room hotter than can be helped. The almost universal practice of the merchant service and Navy now is to circulate the water *through* the tubes. Independently of the particular reason for the choice of the Admiralty, the balance of argument is in favour of circulating the water through the tubes; for when this is the case there is (i.) a larger surface of metal exposed to the hot steam; (ii.) the grease that may be deposited on the tubes is easily removed by working a trifle warm, and using a solution of caustic soda or potash, and if this does not remove it, the deposit being soft does not prevent the tubes from being easily drawn, as is the case when scale from salt-water is deposited on their exterior surface; (iii.) the scale from sea-water, which must be removed mechanically, can be so done without removing the tubes; (iv.) a more extended and complete circulation of the cooling water is possible, and that without risk of air accumulation and without special and expensive diaphragms, etc.; (v.) the condenser is more easily designed, and fits into the general arrangement of most engines, and is smaller, inasmuch as there is no need of an expansion chamber in front of the tubes, as is the case when steam passes through them; (vi.) when it is necessary to examine the packings, or to plug a defective tube, only a water joint is broken; and (vii.) the thin tubes are stronger to resist internal than external pressures, especially of brass.

On the other hand, when the steam passes through the tubes the bulk of the fatty matter is deposited on the front tube-plate, and prevented from coating the tubes; the large flat sides of the condenser are subject to the very slight pressure due to the head of water, and so may be made much lighter; and there is less hot surface exposed in the engine-room. As little or no oil is now used for direct internal lubrication, and such as gets in with the piston and valve-rods is small, and a mineral oil, the argument as regards grease deposit is considerably modified from what it was when animal and vegetable oils were freely used, and even tallow on occasion, in both cylinders and boilers.

Spacing of Tubes, etc.—The holes for ferrules or glands are usually $\frac{1}{4}$ inch larger in diameter than the tubes; but when absolutely necessary the wood ferrules may be only $\frac{3}{32}$ inch thick.

The pitch of tubes when packed with wood ferrules is usually $\frac{1}{4}$ inch more than the diameter of the ferrule hole. For example, the tube being $\frac{3}{4}$ inch external diameter, the ferrule hole will be 1 inch and the pitch of the tubes $1\frac{1}{4}$ inch. In the Navy, with tubes $\frac{5}{8}$ inch diameter and the tube-plates $\frac{7}{8}$ inch to 1 inch thick, the pitch of holes is $\frac{31}{32}$ inch, and in "Destroyers" only $\frac{15}{16}$ inch. In the mercantile marine, with tubes $\frac{3}{4}$ inch diameter and the tube-plates 1 inch thick, the pitch is $1\frac{1}{8}$ inch; for $\frac{7}{8}$-inch tubes, $1\frac{5}{16}$ inch; and for 1-inch tubes, $1\frac{7}{16}$ inch. The tubes are generally arranged zigzag, and the number which may be fitted into a square foot of plate is as in Table xlii.

TABLE XLII.

Pitch of Tubes.	No. in a square foot.	Pitch of Tubes.	No. in a square foot.	Pitch of Tubes.	No. in a square foot.
Inches.		Inches.		Inches.	
$\frac{31}{32}$	184	...	...	...	...
1	172	$1\frac{5}{32}$	128	$1\frac{1}{4}$	110
$1\frac{1}{32}$	161	...	...	...	...
$1\frac{1}{16}$	152	$1\frac{3}{16}$	122	$1\frac{9}{32}$	105
$1\frac{3}{32}$	144	...	...	...	...
$1\frac{1}{8}$	136	$1\frac{7}{32}$	116	$1\frac{5}{16}$	99

The Body of the Condenser.—The surface-condenser was generally in the form of a cylinder or a rectangular parallelepiped, and sometimes a flattened cylinder; the first and last forms are the best suited when weight is a great consideration, and the second and most convenient when space is of first importance; the cylindrical form is by far the cheapest, as the patterns are very simple—the body, when not made of steel, copper, or brass sheets, being *struck out*; the covers, tube-plates, and corresponding flanges can all be faced in a lathe; this form also is by far the lightest, for the two reasons, that the circular plate is the form giving the minimum perimeter for a given area, and consequently a minimum barrel, and that the cylindrical form for either internal or external pressure is the strongest, so that the thickness of metal will be the minimum. A modification of the cylindrical form possesses many of these advantages.

The waterways, or chambers at each end of the condenser, are sometimes cast with it, and sometimes cast separately; in the latter case there is an additional joint, but this is mitigated by its also forming the tube-plate joint; in the former case there is only one joint less at each end through which air can leak, but the plates are more troublesome to fit, and, except in the case of the cylindrical form, the tube plate flange is difficult to face.

Great care should be taken in designing a condenser that free outlet is given to any air that may collect near the tubes, and all pockets, where it or dead water could lie, should be avoided, as a few of the tubes may get hot and leak from the above causes.

The Construction of the Surface Condenser is to-day in all naval ships, in express steamers, and even in some cargo ships of steel sheets or plates from $\frac{1}{4}$ to $\frac{1}{2}$ inch thick in the body, and of cast iron or brass in the waterways and parts exposed to sea-water in naval ships, and of cast iron in the mercantile marine where weight is not of paramount consideration. It is recognised now that there is no corrosive action on the body exposed to steam and condensed water, since the use of sea-water as the supplementary feed has been given up, and brass in the waterways can be protected by zinc slabs, or even by a few steel studs; cast iron also is easily protected by zinc slabs. The condensers of such craft as destroyers, where the last ounce must be saved, should be of sheet brass, cylindrical in form, and of sufficient thickness to withstand atmospheric pressure and its own weight. With such condensers, as also with the brass waterways, the mudhole and peephole doors may be of cast iron or steel with advantage, as then they form the protectors against corrosion of the brass from sea-water. and can be easily and cheaply replaced.

The shape of the modern condenser is either as an approximation to a triangle, or what is known as heart-shaped in transverse section, as shown in fig. 118. The cylindrical condenser is generally fitted with baffles of some kind, although with the modern entering in so much enlarged, especially as it is for turbines, the need to protect the upper rows of tubes from the pulsating impact of the steam has almost ceased to exist. With the ordinary rectangular or nearly rectangular condenser of the mercantile marine and the circular section exhaust pipe there should still be this screen, as well as means for draining the condensed water when formed without further contact with other tubes.

TABLE XLIII.—TRIALS OF I.J. BATTLESHIP "IBUKI"—CURTIS TURBINES.

Date.	Kinds of Trial.	Barometer in Inches.	Temperature (Deg. Fahr.). Super Heat after through Regulating Valve. Starboard.	Port.	Main Condenser. Sea Water F°. Inlet.	Outlet.	Condensed Water, F°.	Feed-Water after through Feed Heater.	Mean Vacuum in Main Condensers at Top.	Bottom
Aug. 12, 1909,	Full power	29·995	21·8	22·9	74·6	97·7	90·9	131·11	26·33	28·42
Aug. 7, 1909,	4/5 ,,	29·990	15·0	10·3	71·2	91·4	86·0	132·75	26 90	28·25
July 31, 1909,	3/5 ,,	29·900	9·8	3·9	75·0	93·4	88·5	151·6	27·10	27·90
July 26, 1909,	2/5 ,,	30·300	1·3	2·0	73·2	90·8	86·7	136·8	27·80	28·50
July 24, 1909,	1/5 ,,	30·320	3·1	4·3	78·2	92·3	88·7	145·2	28·03	28·32

Date.	Boiler. S.H.P. per Sq. Foot of Grate.	Water Rate of Main Engine per S.H.P. Not Corrected to Contract Conditions. (Per hour.)	Corrected to Contract Conditions. (Per hour.)	Main Condenser. Condensed Water per Sq. Ft. of Cooling Surface per Hour in Lbs.	Circulating Sea Water per Lb. of Steam in Lbs.	Main Engine. Bucket Speed in Feet per Sec.	Steam Velocity in Feet per Sec (1st Stage.)	Efficiency of Turbine. Per cent.
Aug. 12, 1909,	16·45	15·050	13·77	11·505	44·296	157·3	2,142·5	52·45
Aug. 7, 1909.	12·80	15·686	14·76	9·099	39·685	147·7	2,330·0	48·95
July 31, 1909.	9·72	16·505	15·62	7·253	41·113	136·2	2,379·0	46·30
July 26, 1909,	9·23	18·652	17·75	5·038	31·056	118·4	2,650·0	40·70
July 24, 1909,	9·05	21·074	20·57	3·065	30·946	95·6	2,943·0	35·10

The Stiffening of Flat Surfaces should be by means of ribs cast with it when of cast iron or cast brass, and 25 thicknesses apart for cast iron and 40 for tough brass and sheet steel and brass. which should have angle or T-iron stiffeners riveted on. Malleable flat surfaces can be stiffened by corrugations formed with it about 35 thicknesses pitch if fairly deep; if the corrugating is light, the pitch should be much less.

Quantity of Cooling Water.—The necessary amount of circulating water may be calculated in the same way as that for injection water (see p. 345), on the principle that the exhaust steam has a certain quantity of heat which is to be expended in raising a mass of sea-water of a certain temperature to nearly that corresponding to that in the condenser. The quantity of sea-water will, therefore, depend on its initial temperature, which in actual practice may vary from 40° in the winter of temperate zones to 80° of the West Indies and other subtropical seas. In the latter case, with a vacuum of 28 inches, a pound of water requires only 20 thermal units to raise it to 100°, while 60 are required in the former. From this it is seen that the quantity of circulating water required in the tropics is three times that of the North Atlantic in the spring of the year.

As before, let T_1 be temperature of the steam on entering the condenser, and L the latent heat: T_0 the temperature of the circulating water, and Q its quantity; T_2 the temperature of the water on leaving the condenser, and T_3 the temperature of the hot-well.

The heat to be absorbed by the cooling water is now $(T_1 + L) - T_3$; and this amount of heat must be equal to $Q\,(T_2 - T_0)$. Hence,

$$Q = (T_1 + L) - T_3 \div (T_2 - T_0).$$

Or

$$Q = \frac{1{,}114 + 0{\cdot}3\,T_1 - T_3}{T_2 - T_0}.$$

Example.—To find the amount of circulating water required by an engine whose steam exhausts at 8 lbs. pressure absolute, the temperature of the sea being 60°, and (2) the amount required when the temperature of the sea is 75°. The temperature of the hot-well to be 120°, and that of the water at the discharge 100°. Vacuum 26 inches. The temperature corresponding to 8 lbs. is 183°.

(1)
$$Q = \frac{1{,}114 + 0{\cdot}3 \times 183 - 120}{100 - 60} = 26{\cdot}22.$$

That is, the water required is 26·22 times the weight of steam.

With the jet condenser under similar conditions the quantity was only 17·48 times (*v.* p. 346).

(2) When the sea is at 75°

$$Q = \frac{1{,}114 + 0{\cdot}3 \times 183 - 120}{100 - 75} = 41{\cdot}95 \text{ times}.$$

It is usual to provide pumping power sufficient to supply 30 times the weight of steam for general traders, and as much as 40 times for ships working in subtropical seas. As will be shown in another chapter, if the circulating pump is double-acting, its capacity may be $\frac{1}{53}$ in the former, and $\frac{1}{42}$ in the latter case of the capacity of the low-pressure cylinder.

Table xliv. shows the least weight of cooling water required in practice per pound of steam entering the condenser at a pressure of 12 lbs. absolute. Modern air pumps can maintain a vacuum of 29 inches, but it will be seen by this table that the quantity of cooling water required for it at a temperature of 70° is enormous, and with water at 80° is impossible.

TABLE XLIV.—Ratio of Cooling Water to Steam Condensed.

Vacuum,	. Ins.	25·0	25·5	26·0	26·5	27·0	27·5	28·0	28·5	29·0	29·3
Sea water,	. 50° F.	13·9	14·7	15·4	16·4	18·2	20·1	22·8	28·0	40·4	65·0
,,	. 60° F.	16·0	17·1	18·1	19·5	22·1	24·8	29·0	38·0	64·3	158
,,	. 70° F.	18·9	20·5	21·8	24·0	27·9	32·3	40·0	66·3	156	..
,,	. 80° F.	23·1	25·5	27·6	31·0	37·9	46·3	63·0	133	..	..

The highest possible vacuum with the temperature of cooling water at 80° is 28·8; and the ratio no less than 270. The highest vacuum when the water is 70° will be 29·1 inches, and the ratio then 220. In the temperate zone with sea-water at 60° a vacuum of 29·4 can be maintained by passing 280 times the weight of steam condensed.

In every-day practice, and the condenser not too clean, the quantity of water required may easily be 10 to 20 per cent. more than that given in this table.

Passage of Circulating Water.—The water must be caused to pass over a sufficient amount of surface to become duly heated, if the minimum quantity is to be used. In practice it should travel at least 20 feet lineally through the tubes before leaving the condenser; if this cannot be arranged, then it must remain longer in contact with the surface. Hence, in small condensers, where the steam is outside the tubes, the water circulates only twice through them at a slow pace; in larger condensers it may circulate twice through long tubes, or three or four times through shorter tubes at a higher velocity, due to the larger quantity of water. To obtain the best results the velocity of flow should be not less than 200 feet per minute when the sea-water is at 60°, and 300 feet when at 75°. When the water circulated outside the tubes, it was necessary to fit baffle-plates, to divert the water and prevent it from taking the shortest course from the inlet to the outlet, and also to prevent the hot water from accumulating at the top part and the cold water from remaining in the bottom. Some skill and ingenuity were often required to fully overcome such difficulties. Condensers are, however, seldom or never made now with the water outside the tubes.

Where reciprocating pumps are employed, the circulating water should be admitted to the condenser direct from the sea, and *pumped from it* through an outlet above the level of the top row of tubes; the pressure in the condenser does not then exceed that due to the head of water, and little or no shock is given to it by the varying velocity of the circulating pump, as is the case when the water is forced through. When a centrifugal pump is used, it should deliver into the condenser direct from the sea.

The Size of the Circulating Pump Pipes may be calculated in the following manner, where C is the consumption of steam per I.H.P. per hour, K the ratio of cooling water to steam condensed. Then :—

$$\text{Total quantity of cooling water in pounds per minute} = \frac{C \times K \times \text{I.H.P.}}{60}.$$

If the maximum flow of water be taken as an average of 600 in temperate zones, where K does not exceed 64, and 800 in the tropics when K may be 130. The weight of a cubic foot of sea-water being taken at 64 lbs.

$$\left.\begin{array}{r}\text{Then volume of water}\\ \text{used per minute}\end{array}\right\} = \frac{C \times 64 \times \text{I.H.P.}}{60 \times 64} \text{ cubic feet}$$

$$= \frac{C \times \text{I.H.P.}}{60} \text{ cubic feet.}$$

$$\text{The area of pipe section} = \frac{C \times \text{I.H.P.}}{60 \times 600} \text{ square feet in temperate zone.}$$

$$\text{,, \quad ,, \quad ,,} = \frac{C \times \text{I.H.P.}}{30 \times 800} \text{ square feet for tropical work.}$$

$$\text{The diameter of the pipe} = \sqrt{\frac{C \times \text{I.H.P.}}{36{,}000 \times 0{\cdot}7854}} \text{ and } \sqrt{\frac{C \times \text{I.H.P.}}{24{,}000 \times 0{\cdot}7854}}$$

$$= \frac{\sqrt{C \times \text{I.H.P.}}}{165} \text{ and } \frac{\sqrt{C \times \text{I.H.P.}}}{137{\cdot}3} \text{ feet.}$$

$$\left.\begin{array}{r}\text{Or, diameter of circulating}\\ \text{water pipes in inches}\end{array}\right\} = \frac{\sqrt{C \times \text{I.H.P.}}}{x},$$

where for European waters $x = 13{\cdot}8$, and for tropical waters $11{\cdot}5$.

Example.—A ship intended to cross the equator is to be of 2,000 I.H.P.; her engines require altogether 16 lbs. of steam per I.H.P. per hour, what size circulating pipes should she have?

$$\text{Diameter} = \sqrt{16 \times 2{,}000} \div 11{\cdot}5 \text{ or } 15{\cdot}6 \text{ inches.}$$

Size of Inlet and Discharge Pipes* should be such that the flow of water through them when the engines are working at full speed does not exceed 700 feet per minute. In temperate zones 7 lbs. per minute of cooling water per I.H.P. is sufficient for a triple-compound engine, and 6 lbs. will do for a quadruple engine or a triple working at a fairly high rate of expansion with 27·5 inches vacuum, but for the same vacuum in the tropics it will require 12 and 10·6 lbs. (*v.* Table xliv.). Hence

$$\text{Area section inlet and discharge} = \frac{\text{I.H.P.}}{C} \text{ sq. ins.}$$

Where C = 32 triple and quadruple high expansion, general,
= 30 ,, ,, moderate ,, ,,
= 26 ,, ,, tropics ,, (if 27·5 vacuum).

Extra Supply Cock.—To provide for the water lost in waste, leakage, etc., it is usual to fit a small cock, through which some of the circulating water may be passed to the steam side of the tubes. The pipe for this should be about one-third the diameter of the main feed pipe, the velocity of flow being nearly nine times that usually provided for in feed pipes. Now that evaporators are in general use, so that the waste can be made up with fresh water. the necessity for such a fitting scarcely exists, but as an emergency provision it may remain. For reserve water such a cock is convenient.

* The flow should not exceed the rates given by this rule :—$v = 350 \sqrt[3]{\text{diameter pipe}}$.

Man-holes and Mud-holes.—A man-hole is necessary for the purpose of admitting men to clean, repair, or tube the condenser, and smaller holes should be provided through which mud, scale, etc., may be scraped out. Peep-holes are sometimes formed in the doors, especially if they are large and heavy, through which the tube ends may be seen and examined. These latter are useful when steam is condensed inside the tubes, to admit the nozzle of a steam jet to wash away deposit.

Drain Cocks should be fitted so that the condenser may be thoroughly drained when not in use.

Testing.—The Admiralty require condensers to be tested to 30 lbs. per square inch before being placed in the ship, and many steamship companies require the same test, while others are content to test with lower pressures. To provide for such loads, the flat surfaces must be stiffened in the same way as laid down for cylinders, and, if flat-sided, stiffened as already prescribed, and, when necessary, tied together by stay bars, etc.

Cementing.—It is a good plan to cover the inside of iron condensers, where there is much wash of condensed water, with a good coating of Portland cement, and under the air-pump and in the pump passages it should be at least a quarter of an inch thick.

Evaporators.—Since the use of steam of 150 lbs. and upwards the extra supply from the sea has been avoided as much as possible, fresh water being carried in tanks or double bottoms; now the employment of "evaporators" is doing away with the necessity for this and providing a long-felt want, and permitting of the use of water-tube boilers and modified forms of ordinary marine types.

CHAPTER XV.

PUMPS.

Air-Pump.—The function of this pump in all condensers is to abstract the water condensed, and the air which was contained in the water when it entered the boiler; and in the case of jet-condensers, it pumps out in addition the water of condensation and the freed air which it contained. Further, it must withdraw all air which leaks into the condenser.

It follows, then, that the size of the air-pump must be calculated from these conditions, and allowance made for the efficiency of the pump; or, what is the same thing, the result thus found must be multiplied by the ratio between what the pump should do theoretically, supposing its action perfect, and what it does actually in practice.

Ordinary sea-water contains, mechanically mixed with it, one-twentieth of its volume of air, when under the atmospheric pressure. Now, suppose the pressure in a jet condenser to be 2 pounds, and the atmospheric pressure 15 pounds, neglecting the effect of temperature, the air on entering the condenser will be expanded to $\frac{15}{2}$ times its original volume; so that a cubic foot of sea-water, when it has entered the condenser, is represented by a cubic foot of water, and $\frac{15}{40}$ of a cubic foot of air.

Now let q be the volume of water condensed per minute, and Q the volume of sea-water required to condense it; and let T_2 be the temperature of the condenser, and T_1 that of the sea-water:—

Then $(q + Q)$ will be the volume of water to be pumped from the condenser per minute, and

$$\frac{15}{40}(q + Q) \times \frac{T_2 + 461^\circ}{T_1 + 461^\circ} \text{ the quantity of air.*}$$

If the temperature of the condenser be taken at 120°, and that of sea-water at 60°, the quantity of air will then be $\cdot 418\ (q + Q)$, so that the total volume to be abstracted will be

$$(q + Q) + \cdot 418\ (q + Q) = 1\cdot 418\ (q + Q).$$

Now, if the average quantity of injection water be taken at 26 times that condensed, $q + Q$ will equal $27\ q$.

Therefore, volume to be pumped from the condenser per minute

$$= 38\ q.$$

Example.—To find the theoretical capacity of a single-acting pump for

* Absolute zero point, or point of no heat, is 461° below the zero of Fahrenheit's thermometer.

an engine using 3 cubic feet of water per minute. and the number of strokes being 40.

$$\text{Volume to be pumped out} = 38 \times 3, \text{ or } 114 \text{ cubic feet.}$$

$$\text{Therefore, the capacity of the pump} = \frac{114}{40}, \text{ or } 2{\cdot}85 \text{ cubic feet.}$$

If the stroke of the pump be taken at 1·5 feet.

$$\text{The area of bucket} = \frac{2{\cdot}85}{1{\cdot}5}, \text{ or } 1{\cdot}9 \text{ square feet.}$$

Example.—To find the theoretical capacity of a double-acting pump for an engine using 10 cubic feet of water per minute, the number of revolutions being 90, and the stroke of the pump 2·5 feet.

$$\text{Volume to be pumped out} = 38 \times 10, \text{ or } 380 \text{ cubic feet.}$$

$$\text{Area of bucket} = \frac{380}{90 \times 2 \times 2{\cdot}5}, \text{ or } {\cdot}844 \text{ square foot.}$$

Of course, were these examples worked strictly, it would be necessary to find the relation between Q and q in each case, instead of assuming it at 26, as has been done.

The foregoing calculations, etc., are for jet-condensers, and based on the supposition that the air-pump also abstracts the condensing water; now in a surface-condenser, not only is the air-pump relieved of this latter duty, but, since the feed-water is condensed steam, and has not had time to absorb much air, it would seem that practically its only function is to draw off the condensed water, and should, therefore, be no larger than a feed pump, as no doubt it might be, did the engine work perfectly; but since ordinary water has to be occasionally admitted to make up for waste, and as slight leakage at the glands and joints very frequently exists, it would be necessary and sufficient to make the air-pump about half the size that would be requisite for jet condensation; but when a surface-condenser was arranged so as to be worked as a jet-condenser, the air-pump had to be large enough to do the work necessitated by this, unless the circulating pump was fitted so as to be worked as an air-pump. A surface-condenser is now seldom arranged as to admit of jet condensation. Too great care cannot be expended on the design of the air-pump for a surface-condenser, as the success of many an engine in the past was marred by a bad vacuum; and doubtless an inch or two of vacuum tells wonderfully on an engine, especially on a compound engine, where the degree of vacuum at any time can be told almost by variation of its speed. Since, however, the surface-condenser was required to work as a jet-condenser, when so fitted, only in cases of emergency, which seldom happen, and when such a case does occur it is not of importance that the engines shall work at the highest efficiency or maximum speed, it seems better so to design the air-pump as best to suit surface condensation, rather than to make it of the larger size, and sacrifice a large amount of work in driving it during the long period, when jet condensation is not necessary. In short, the pump should be of such a size as to produce maximum efficiency during the longest time, and if for a surface-condensing engine, it should be designed for that particular service.

Air-Pumps were originally simple lift pumps, having a valve in the bucket and a delivery or "head" valve at the top, through which was passed the water of condensation, mingled with which was the condensed water and the air or non-condensable gaseous matter passing to the cylinder from the boiler, or that emanating from the water of condensation being previously mechanically mixed or in solution with it. The *modus operandi* was simple; the pump bucket descended to the bottom of the (vertical) chamber, and in so doing expanded such gaseous matter as remained locked between it and the head valve until the pressure was less than that in the condenser, when

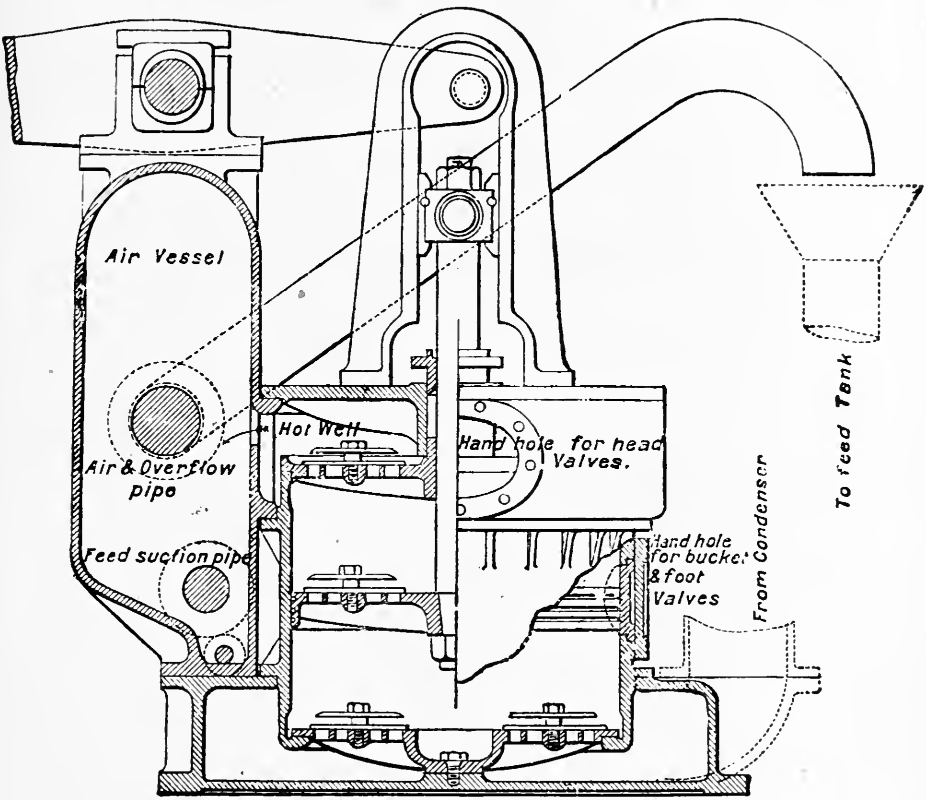


Fig. 122.—Air-pump of Ordinary Type.

there flowed into it the gaseous residuum of the condenser and the water until the bucket ceased to move and the valve closed; the bucket on ascending compressed the gases till their pressure was higher than that of the atmosphere, when the head valve opened, and they together with the charge of water were delivered into and separated at the hot-well.

With the Horizontal Engine the Air-Pumps were usually horizontal and double-acting, so that the bucket no longer had a valve; moreover, sometimes in lieu of a cylindrical pump chamber and piston or bucket, there was a plunger passing through a stuffing-box in the diaphragm separating the back from the front chamber of the pump. Although with a well-designed horizontal pump in good working order quite high vacua were sometimes maintained, as a rule such pumps were not nearly so efficient as a vertical pump. They had the same stroke as the engine, and as the valves were quite apart from the pump, they could be of any reasonable number or size. A return to such pumps is now most unlikely, for, as a fact, even horizontal engines had latterly vertical pumps whenever possible. But some makers of vertical engines have preferred to fit pumps worked direct from the piston, so that their stroke was large and volume clearance small, but the passages through their buckets is likewise small, and consequently a source of danger with a leaky condenser.

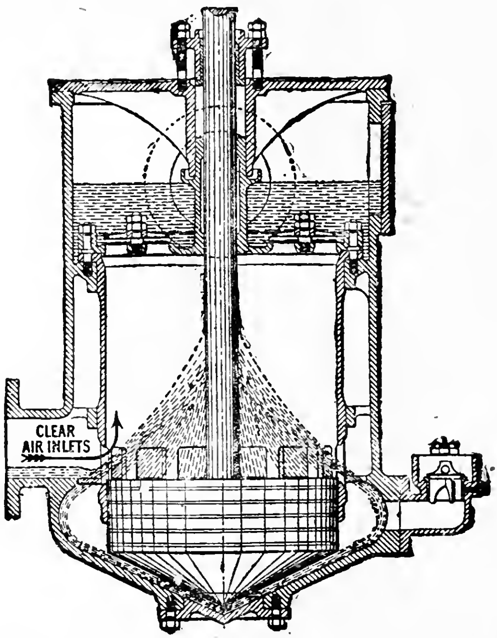


Fig. 123.—Edwards' Air-pump. Bucket at Bottom of Stroke.

The Single-acting Vertical Air-pump (fig. 122), having valves in the bucket as well as foot and delivery valves, is by far the most efficient, and, when possible, was generally chosen. If the rod of such a pump is enlarged or the bucket has a trunk to surround the rod, which is attached to a joint in its centre, it is to a certain extent double-acting, since on the upstroke it causes the chamber in which it works to fill, on the downstroke it displaces, and consequently discharges a volume equal to the volume of the trunk, and on the upstroke discharges the remainder. If the sectional area of the trunk is half that of the bucket, the discharges are equal, and the pump is virtually a double-acting one.

Edwards' Air-pump is an ingenious form of single-acting vertical pump, with valves only at the top to check the discharged water and air from returning to the pump on the downstroke. Fig. 123 shows the general design and action of the bucket or piston of this pump, whereby the water and air are caused to enter the exhausted space between the head-valves and the bucket. These pumps are very simple, have the minimum number of valves, which are easily examined and replaced, work with the least possible resistance and wear and tear, and produce a good vacuum.

The Double-acting Air-pump.—It is not always convenient to have a vertical pump in the horizontal engine, and consequently a horizontal pump was generally employed, and this was almost of necessity double-acting. The bucket in this case works air-tight in a smooth barrel placed beneath the

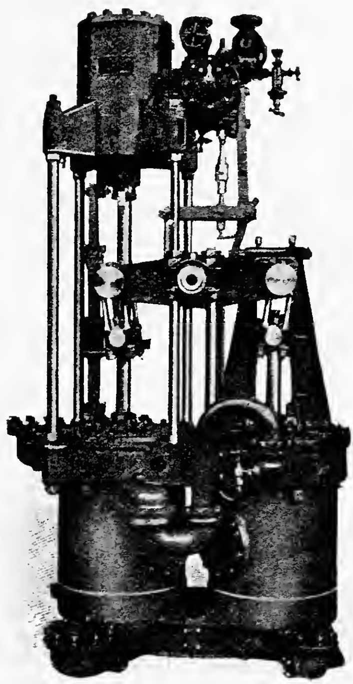


Fig 124.—Weir " Dual " Air-pump (direct-driven).

condenser, and has a set of foot and delivery valves for each end. Sometimes, in lieu of a barrel and bucket, a plunger is fitted, passing through a stuffing-box in the diaphragm-plate dividing the condenser bottom. This latter arrangement generally admits of more room for, and a better disposition of, the valves. Some engineers adopted this form of the horizontal pump for twin-screw vertical engines, and worked it by means of an eccentric formed on one of the crank-arms.

Weir's Dual Pumps are shown in fig. 124, and in the diagrammatic form fig. 124*a* the full working of this ingenious and effective system is clearly seen. There are two pumps—one called the **Wet Pump** draws water and such "**air**" as it can from the condenser in the usual way, and discharges into the hot-well; the other, which may be looked on as the additional or auxiliary, is

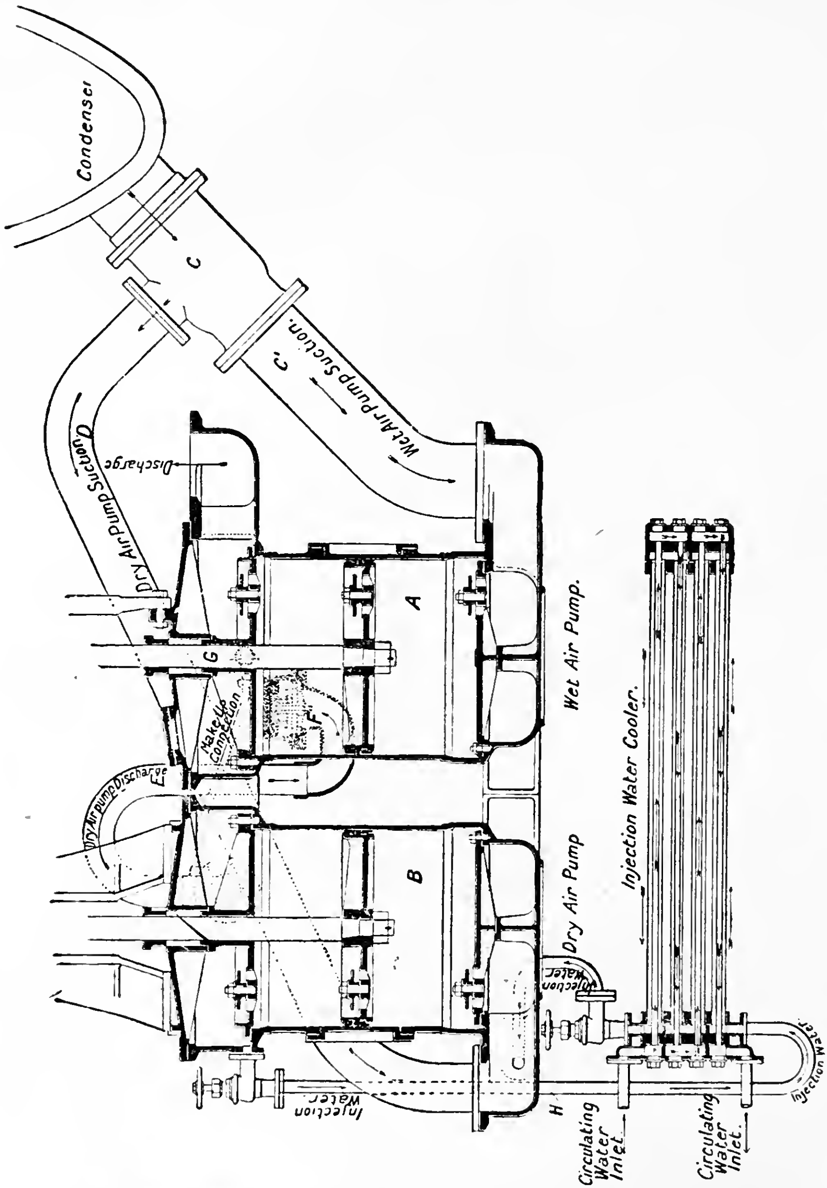


Fig. 124*a*.—Weir's Dual Air-pumps and Cooler Connections (Diagrammatic).

called the **Dry Pump,** inasmuch as it draws no water direct from the condenser, but the gaseous products only; it does not, however, deliver them to the hot-well or other open receptacle, but into the chamber of the other pump between the bucket and the head valves, and then only when the tension there is low for a delivery or non-return valve prevents the contents of the first pump when at high compression flowing into the second. In this way there is maintained a two-stage delivery of the gaseous products, and as the two pumps are worked from the opposite ends of a rocking lever, the dry pump is delivering only when the piston of the other is falling and expanding the contents of its chamber. The result is that the contents of the dry pump are delivered at quite a low pressure, and the whole compressed and discharged to the hot-well by the wet pump. The pressure in the wet pump under these circumstances is never very low, so that any air it gets must be mechanically mixed with the water, which, under the action of gravity, flows into the wet pump. It is obvious that the work of the two pumps is quite different, while there is active co-operation between them; also that their efficiency will be high, and the air leakage not nearly so serious as those in pumps where the difference in pressure at every point is greater. But the second or **dry pump** is somewhat misnamed, inasmuch as it is supplied with water with which to lubricate the moving parts as well as seal them against air leaks, and inasmuch as that water is very cold, it serves the more important function of being the means whereby the gases are cooled and the pump kept cool, so that the low vacuum desired in it may be obtained. This water is cooled in the pipe coils shown, by means of sea-water, and is really a small auxiliary condenser, which serves the same purpose as the few drowned tubes in the Morison condenser (fig. 118).

The following is Mr. Weir's own description of the dual pumps and their action:—

Fig. 124*a* shows in a diagrammatic form the arrangement of surface-condenser, "dual" air-pump, and injection water cooler. In all cases the pump A or wet pump is situated below the steam cylinder, as this pump is the only one which works under any considerable load, the dry pump B is driven by the beam and links in the usual manner. One connection C is made to the condenser, but a branch pipe D is led to the dry pump, the connection being made in such a manner that the water will all pass by C_1 to the wet pump. Both pumps are generally of the three-valve marine type, but in certain cases the dry pump may be of the suction valveless type.

The first and most important difference from an ordinary twin pump consists in the separate suction to each pump, then in the dry pump discharging through the return pipe E, through a spring-loaded valve F, into the wet pump at a point below its head valves. The next point concerns the supply of water to the dry pump for water sealing, clearance filling, cooling, and vapour condensing. When starting the pump the filling valve G must be opened for a minute or so to enable the vacuum to draw in a supply from the hot-well of the wet pump. The valve is then closed, and the water passes from the hot-well of the dry pump by the pipe H to the annular cooler, through which a supply of cold sea-water circulates, and after being cooled passes into the suction of the dry pump, then passing through the pump it becomes heated and again passes to the cooler, and so on in a continuous closed circuit, any excess passing over the pipe E to the wet pump. The spring-loaded valve F is adjusted to maintain about 20 inches vacuum in the dry pump hot-well when the condenser is working at 28 inches vacuum, and this 8 inches difference of pressure is sufficient to cause the water to overcome the cooler friction and pass into the suction, and at the same time never allow any direct air connection between the dry suction and discharge.

Air-pump Indicator Diagrams, shown on fig. 125, explain the action of each kind of pump now used on shipboard for extracting air from a condenser, assuming that the valve resistance is nil. The pressure in the condenser is represented by L C, that of the external air O J, and the head at delivery above that pressure A H. If no water is passing, A H will be nil.

A J is the equivalent clearance above the bucket, and D K the full chamber. Taking first a simple pump like the Edwards, and suppose the bucket at the top of its stroke A, and the clearance space charged with air of pressure O J. On descending this is expanded, as shown by the curve A B D, so that the pressure in the chamber at the end of the stroke would be L D, while that in the condenser is L C. But before reaching the point D, the piston passes the openings in the barrel at M, when the inflow from the condenser takes place from M to C. The bucket then begins to rise, and the compression of air takes place on the curve C G, until the pressure is equal to that in the hot-well, when it commences to discharge till the pressure is at H. If, however, the pump has valves in the bucket and foot-valves without clearance under the bucket, then the following cycle is followed. Expansion

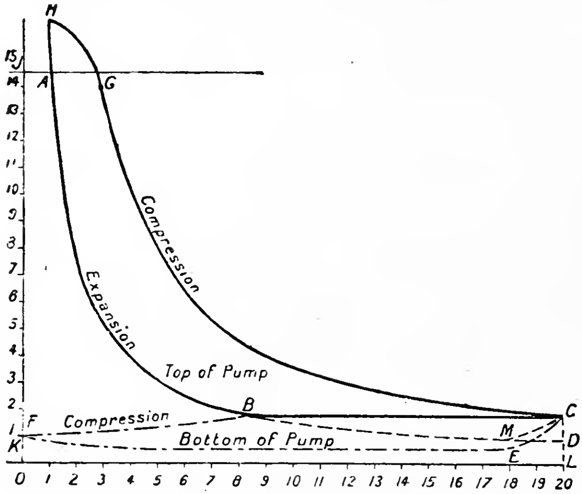


Fig. 125.—Indicator Diagram of Air-pumps (Theoretical).

takes place above the bucket as before, but now compression below it also takes place until the pressure below is equal to that above the bucket—that is, at the point B, where the expansion curve A B intersects the compression curve F B, during the remainder of the stroke the pressure above and below is L C; the valves close, and the bucket rises, compressing the air above it on the curve C G as before; but now expansion takes place below, following the curve C E until the pressure is so much below that in the condenser (O F in this instance) that vapour and air flow into the chamber, and finally fill it at a pressure O F, and so on, as before.

It will be seen that, with the same clearance on top of the bucket, the Edwards pump cannot produce so good a vacuum as the common one with foot-valves. As a matter of fact, however, the Edwards has naturally smaller clearance space, and both kinds of pump have virtually little or no real clearance, as the space is filled with water. The water itself, however, has some air in solution, and probably such as fills the space is frothy, there will be thus a modicum of gaseous matter to expand, and the cooler the chamber is the less will the expansion be. Hence the supply of water to seal the pump and fill the spaces should be cold and as free from air as possible.

The Efficiency of Air-pumps.—The high efficiency of the single-acting vertical pump is due to the certainty of its action in taking the water, etc., through the bucket-valves, and to the valves from their position so readily closing when required; there is also time for the water to drain into the bottom of the pump during its upstroke, and collect there ready for the bucket when it descends. Moreover, as the flow is always in one direction, the velocity of flow is not checked by diversion. The water always lies on the valves so as to render them air-tight, and there is very little clearance space, as a rule, between the foot and bucket valves, and between the bucket and head valves, and what there is contains water, and, therefore, virtually there is no clearance at all for air.

The low efficiency of the double-acting horizontal pump is caused by the reverse of some of the above conditions, especially by the failure of the valves in closing, and to the large space between the foot and delivery valves, also by leakage at the gland of the rod, and past the bucket, which is only lubricated by the water on the bottom, and in no small degree by the ever-changing direction of flow. The latter defect is proved by stopping one end of the pump, when it is often found (especially in the case of badly designed pumps, etc.) that the vacuum is not very materially altered, although sometimes improved. The foot valves are sometimes kept covered with water by allowing the water to pass back again through a pipe from the hot-well to the pump-chamber.

If an Air-pump has no Foot Valves, or, like the Edwards pump, the clearance real or virtual must be such that there is a vacuum in the pump chamber superior to that in the condenser an appreciable time before the end of the stroke, so that some of the gaseous contents of the condenser may enter it. If, therefore, p_c is the pressure to be maintained in the condenser, x the clearance fraction of stroke S, the atmospheric pressure, say, 15 lbs., and the weight of the valves and water "head" and resistance equal to another pound, then the pump must compress on the upstroke to a pressure of 16 lbs. to deliver its contents, and on the closing of the valves the pressure of gases remaining locked in the clearance will be 15 lbs.

The clearance (virtual) must not be less than $\frac{p_c}{15}$ S to effect any inflow at all, but as this inflow must commence before the end of the stroke, and in the Edwards pump there must be a much lower pressure than in the condenser to carry away considerable quantities of vapour, x must be much less than $\frac{p_c}{15}$, and should be $\frac{p_c}{20}$ at most.

During the down-stroke expansion of the gases takes place until the pressure is less than that below the piston or bucket, and from that point to the end of the stroke the pump is being charged from the condenser through the bucket valves. The pressure of the gases in the pump is then p_c, which on compression must be compressed to 16, hence the ratio of pump chamber to its clearance will be $16 \div p_c$ when delivery takes place, and since this must be before the end of the stroke, it follows that x must be less than $\frac{p_c}{16}$.

If, therefore, a pump is to maintain a vacuum of 28 inches at all, its clearance must be less than $\frac{1}{16}$, and should be $\frac{1}{20}$ to $\frac{1}{25}$.

For so high a vacuum as 29 inches it must be even higher, and not less than $\frac{1}{32}$, and should be, if possible, $\frac{1}{40}$.

Since, however, the pump is always charged with water, so as to seal the bucket, the valves and other sources of air leak, and generally fills the clearance space, the virtual clearance in a properly designed pump is exceedingly small.

But to form and maintain in the pump chamber a vacuum under 28 inches, the water entering it must never exceed 95° F., and for 29 inches it should be under 78° F. If, therefore, it is desirable to obtain the condensed water as warm as possible with such high vacua, there should be two pumps, one dealing with the water, the other with the gases, and a special supply of cold water to fill the clearance and seal the leaks, as done virtually by Mr. Weir.

If an Air-pump has Foot-valves the conditions obtaining at the lower end of the pump differ from the foregoing, for on the down-stroke the attenuated gases remaining in the pump undergo compression at the same time that the fragment of compressed gases above the bucket are expanded, as already described; when the pressure below is slightly in excess of that above the bucket valves lift and remain open to nearly the bottom of the stroke; when they close the pump is charged with gases at the pressure to which they were compressed, and the further compression of them takes place until the 16 lbs. pressure is attained, when discharge into the hot-well takes place. On this upstroke, however, expansion of the gases between the bucket and the foot valves takes place, and continues till the pressure is somewhat below that of the condenser, when inflow from it commences, so that at the end of the upstroke the pump chamber is filled with gas at a pressure very nearly equal to pc.

Diagram fig. 125 shows the cycle of the top and bottom of an air-pump with foot valves, and the plain dash lines indicate what happens when there is no foot valve. The diagram is constructed on the assumption that the virtual clearance at both top and bottom is equal to 5 per cent., or $\frac{1}{20}$ the stroke, and that expansion and compression follows the law pv = constant, which is sufficiently correct for the purpose here. It is also assumed that there is no resistance to inflow from the condenser or through the bucket.

In the case of the single-acting pump working under these conditions, it will be seen that the highest vacuum will probably be about 27, and by the diagram is shown as 26·5, whereas with the foot valves fitted it is 29 inches in the pump, and 28 in the condenser. It is likewise obvious that, if the virtual clearance can be kept down below 5 per cent., which is really not difficult, even the Edwards or the common pump without a foot valve can maintain high vacua.

The resistance of the pump in question, due to compression of gases, is equal to a mean pressure of 4·92 lbs. per square inch of bucket on the upstroke. In the downstroke the pressure above the piston is greater than that below it for 45 per cent. of the stroke, and practically nothing for the remaining 55 per cent. The *weight* of the bucket and the water helps on the downstroke as much as it resists on the up. But the condensed water has to be lifted a few feet, and there is resistance due to friction. For calculation of power, a mean pressure during the upstroke of 6·5 lbs. will be sufficient, but it must not be overlooked that during about 8 per cent. the resistance is from 15 to 16 lbs. per square inch, and the ratio of maximum to mean resistance is, therefore, as much as 2·46, and has to be overcome by motor or steam piston employed to work the pump. Unfortunately, the maximum resistance comes at the termination of the stroke, so that one cannot fit a cylinder with an early cut-off to the steam. The only relief so far provided is by working the two pumps from the opposite ends of a rocking lever, or by having three pumps worked from a three-throw crank-shaft having the cranks at 120° apart. The ratio of maximum to mean is then considerably reduced, and the work of a motor comparatively easy. This latter is the system generally followed in electric generating central stations (fig. 126). On shipboard such an installation might be

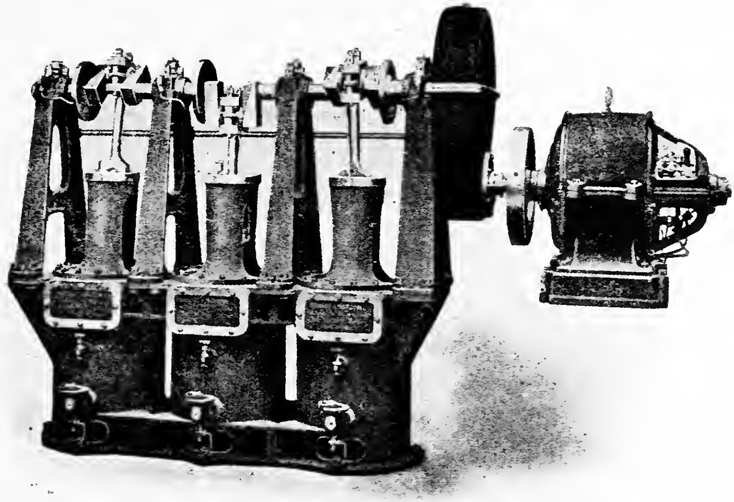


Fig. 126.—Motor-driven Air-pumps (Edwards)

driven by a low-pressure turbine taking its steam from the centrifugal pump cylinder exhaust, or from the feed-pump when operated by a cylinder, as in the Weir type.

Size of Air-pump.—The capacity of the air-pump should be calculated from consideration of the conditions under which it is to work, and by the rules given in this and the preceding chapter, and suited to practice by an allowance made for the efficiency of the pump. If the pump is single-acting and well-designed, and is working under favourable conditions, its efficiency may be taken at 0·6 ; and if the reverse of this 0·4 ; generally its efficiency is about 0·5, so that the size in practice should be double that given by theoretical calculation. The efficiency of the double-acting pump varies from 0·5 to 0·3, and generally is not more than 0·35 ; its size for good working should be nearly three times that given by theoretical calculations

Hence, when the temperature of the sea is 60°, and that of the (jet) condenser is 120°, Q being the *volume* of the cooling water, and q the *volume* of the condensed water in cubic feet, and n the number of *strokes* per minute.

$$\text{The volume of the single-acting pump} = 2 \cdot 74 \left(\frac{Q + q}{n}\right).$$

$$\text{The volume of the double-acting pump} = 4 \left(\frac{Q + q}{n}\right).$$

The Capacity of Air-pumps for a surface-condenser is not easy to determine, inasmuch as, if the condenser is tight and the stuffing-box packings of the main and auxiliary engines in good working order, they have little work to do beyond removing the condensed water after a vacuum has been established. Quite a small pump, therefore, should suffice for the running of an engine when in good working order. It is true that at starting such a small pump will take longer to exhaust the condenser and form a good vacuum, but the vacuum, when formed, will be maintained at less cost than with a bigger pump. But in every-day practice it would be found that with very small air-pumps there are considerable, though only temporary, variations in the vacuum due to leakages of the glands of the L.P. members, to admission of fresh make-up feed-water or to leakage from inattention to the steering engine, or some other auxiliary exhausting to the condenser. If, therefore, the air-pump is worked by the main engine, it must be of sufficient size to form a good vacuum in a short time, and to restore it quickly if temporarily reduced. It should be remembered also that the pump which is barely sufficient for that purpose may be ample at the reduced speed of service, inasmuch as the capacity of the pump is falling off only in direct proportion to the number of revolutions, while the steam consumption is being reduced as the cube of the revolutions, so that at $\frac{3}{4}$ speed while the capacity is only 25 per cent. less the steam condensed is 58 per cent. less. On the other hand, it is quite likely that the pump is not so efficient at the lower speed, due to air leakages in it and at the glands of the main engines. If the air-pump is driven by its own independent steam engine or motor, it may automatically work at the lowest speed compatable with good vacuum, and, therefore, this system is the better one for all installations of large reciprocators, notwithstanding that power derived from them is generated much cheaper than is possible with any auxiliary. With turbines the separately driven air-pump is a necessity.

The Air-pump for a Jet-Condenser could be calculated from the conditions under which it was to work and on first principles. It was, however, the common practice of the early marine engineers to make the vertical single-acting air-pumps of paddle and geared-screw engines with a diameter half that of the steam cylinder, and a stroke half that of the engine, so that the rule really was—

Capacity of a single-acting air-pump = one-eighth that of the steam cylinder.

With the advent of the horizontal engine, and its double-acting pump, engine builders varied in their practice somewhat, but on the average the—

Capacity of horizontal double-acting air-pump = 0·09 the capacity of the steam cylinder.

The diameter being about 0·3 that of the engine piston, and the stroke the same as it was driven direct from it.

The Air-pump of Surface-condensers was made at first almost as large as that of a jet-condenser partly for fear of leakage, but chiefly because these early surface-condensers were fitted so as to be worked on the jet or direct-contact principle. Confidence in the surface-condenser, begotten from experience, has caused all these unnecessary appurtenances and provisions to be swept away, and with them went the big air-pump, so that now, with pumps operated by the main engines, their capacity when single-acting and vertical is very much less than formerly. Hence

The capacity of S.A. air-pump = 0·04 that of the L.P. cylinder.

That is, the ratio of L.P. to pump is 25.

For express steamers making short voyages, such as crossing channels. or trips along rivers and coasts, where high speed must be attained quickly and maintained for short periods, it may be 21. For cargo steamers* or passenger steamers making long voyages there is no need of so large pumps, and the ratio in them may be as high as 30.

Air-pumps operated by Independent Means should be such that, when working at full speed, they can quickly form a high vacuum, and when formed may slow down to that only necessary to maintain it and carry away the condensed water. The guide, to arrive at their proper capacity, as really also for determining that of the engine driving them, is based on the quantity of water consumed. Under good conditions a vacuum of 26 inches can be maintained by a pump whose capacity is equal to 0·3 cubic foot for each pound of water condensed and passing through it per stroke for general working; but to provide for casual leakages 0·5 to 0·6 cubic foot is a more satisfactory allowance, while for higher vacuum, say 28 inches, 0·8 cubic foot should be provided, and for higher vacua and economic engines as much as 1 cubic foot will not be too much for rapid formation and steady maintenance.

Example.—An engine has a 50-inch L.P. cylinder, and when working at 100 revolutions per minute develops 1,200 I.H.P., condenses 15 lbs. of steam per I.H.P. per hour; the stroke of the piston is 3 feet, and that of the lever-driven air-pump is 1 foot, what diameter should it be for a vacuum of 28 inches?

$$\text{Here the consumption per stroke} = \frac{1{,}200 \times 15}{60 \times 100} = 3 \text{ lbs.}$$

$$\text{The capacity of the pump} = 3 \times 0{\cdot}8, \text{ or } 2{\cdot}4 \text{ cubic feet.}$$

The diameter is, therefore

$$\sqrt{\frac{2{\cdot}4 \text{ square feet} \times 144}{0{\cdot}7854}} = 21 \text{ inches.}$$

If there had been three self-acting pumps independently driven at 150 revolutions with a stroke of 9 inches—

$$\text{The capacity of each pump} = \frac{2{\cdot}4 \times 100}{3 \times 150} = 0{\cdot}533 \text{ cubic foot.}$$

$$\text{Area of each bucket} = \frac{0{\cdot}533}{0{\cdot}75}, \text{ or } 0{\cdot}71 \text{ sq. foot, or } 102 \text{ sq. inches.}$$

The diameter of each is, therefore, 11·5 inches.

* N.E. Coast Inst. E. and S. recommend 20 when a steam ejector is fitted to the condenser

The stroke of the vertical air-pump is usually from one-third to one-half that of the engine, but it should be such that the velocity of bucket does not exceed 300 feet per minute, and for continuous running 275 feet.

The vacuum in the condenser is liable to be spoiled by air leakage at the expansion joint of the exhaust pipe. It is usual to provide for the expansion of this pipe, which, in the Navy, is of brass or copper, by fitting a stuffing-box, gland, etc., on the condenser or cylinder into which the pipe end is free to work; unless this box is deep, and has a good taper at the bottom, and the part of the pipe fitting in the stuffing-box of brass and turned to fit, it is difficult to keep it quite tight. Some engineers find it better to fit a brass or copper *bellows* joint. In compound engines some trouble is often experienced from leakage through the drain cocks of the low-pressure cylinder, and since, when the engine is working below full speed, the pressure in this cylinder is less than that of the atmosphere as a rule; to prevent this, pipes should be led from them to the condenser.

The Function of a Perfect Air-pump System is to effect. at the cost of a minimum expenditure of energy:—

(*a*) The highest possible degree of rarefication of the vapour space of a condenser under all conditions of air leakage likely to be met with in practice.

(*b*) The extraction from the condenser, without thermal loss, of the water resulting from the condensation of the steam.

In the Kinetic System the air and non-condensable gases are removed from the condenser by the action of a steam jet followed by a special system of water jets known as the "Kinetic Ejector" (*v.* fig. 127).

Jets of water moving at a high velocity through suitably shaped orifices have been used for many years for the purpose of producing a partial vacuum for various purposes, including the rarefication of the vapour spaces of condensers.

A water jet has of itself, however, a relatively low capacity for assimilating and withdrawing air; but the air-withdrawing capacity of any water jet device is greatly increased if the air to be extracted is previously mixed with steam.

In the case of the kinetic plant the steam is introduced into the air suction pipe through a high-velocity nozzle, thus entraining the air and intimately mixing with it. This steam is condensed on the secondary sprays of the kinetic ejectors, and the resulting liquid carrying with it all occluded air and gases is ejected to the atmosphere by the main water jet.

The steam jet is fed with live steam in the case of a condenser fitted with electrically-driven pumps, or with exhaust steam at a pressure of about 20 lbs. above the atmosphere, if this be available. The quantity of steam required varies according to the amount of air leakage and the size of the engines, but from $\frac{1}{2}$ to $\frac{3}{4}$ of 1 per cent. of the steam to be condensed may be taken as an average figure for installations of considerable dimensions.

The water for the kinetic jet is that of condensation which has already been removed from the condenser, and the whole of the latent heat contained in the steam used in the steam nozzle is absorbed by this water, which is subsequently discharged to the feed tank at a correspondingly higher temperature than when it left the condenser.

The water of condensation is withdrawn from the condenser and discharged against the pressure of the atmosphere by the action of two pumps of the centrifugal type, known as the "head" pump and "pressure" pump respectively. The "head" pump works under the condenser pressure both on the suction and on the delivery sides, and is designed so that it will pass the required quantity of water with an extremely low head on the suction side of the pump. The water is discharged from this pump into a stand pipe or receiver, which provides a natural head of water on the inlet side of the "pressure" pump, by which the water is finally ejected against atmospheric pressure. This arrangement makes it possible to place the pumps only a few inches below the level of the condenser bottom, and to maintain a perfectly regular discharge at all loads, the amount corresponding to the quantity of steam condensed.

Ordinary centrifugal pumps have been successfully used for the withdrawal of the water of condensation from condensers both of the surface and jet types, but for satisfactory operation these pumps must invariably

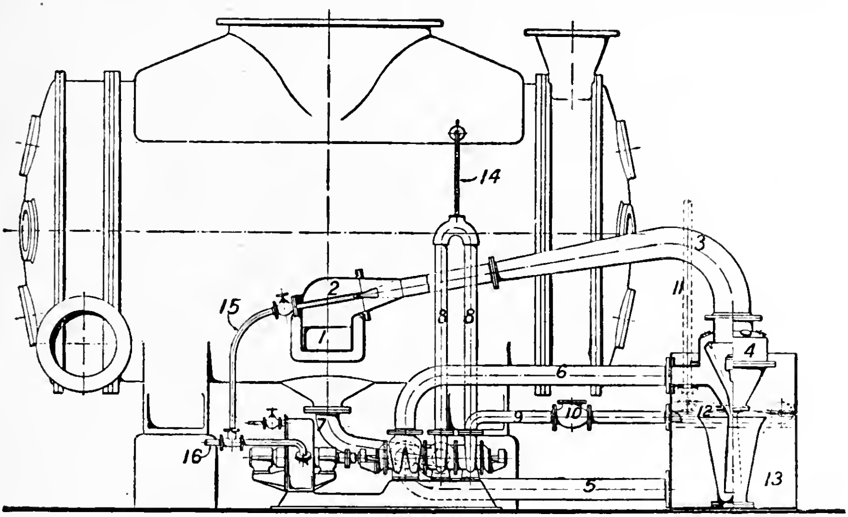


Fig. 127.—Condenser with Kinetic Air-pumps.

1. Air suction orifice on condenser.
2. Exhaust steam jet.
3. Air-pipe to kinetic ejector.
4. Kinetic ejector.
5. Suction pipe to kinetic pump.
6. Kinetic pump to discharge pipe.
7. Condensed water pipe to head pump.
8. Stand pipe between head and pressure pumps.
9. Pressure pump discharge to tank.
10. Non-return valve.
11. Feed-water delivery pipe.
12. Float-controlled feed delivery valve.
13. Kinetic tank.
14. Pressure equalising pipe.
15. Exhaust steam to jet.
16. Surplus exhaust steam

be placed at a considerable distance below the level of the condenser, a condition which is difficult, sometimes impossible, of attainment on shipboard.

For marine use single centrifugal pumps are apt to work very irregularly, due to changes in the relative levels of the condenser and pump consequent on the motion of the vessel.

It is claimed for the kinetic plant that the vertical stand pipe between the "head" and "pressure" pump maintains a practically constant head on the latter irrespective of the motion of the vessel, and that the energy lost from the system in a normally-designed installation does not exceed 0·0003 (or three ten-thousandths) of the energy developed by the total steam which is condensed. This being so, the thermal efficiency of the kinetic system of air and water-extracting pumps is excellent.

It will be noted that the whole of the energy in the steam jet, also that required to drive the air and water-extraction pumps, is returned to the system, except that required for the extraction of the water, and that for the compression of the extracted air against atmospheric pressure.

With these exceptions, which have been proved by experiment to be practically negligible, the whole of the energy expended reappears in the form of heat in the kinetic tank, whence it is returned to the boilers, subject, of course, to such small losses as arise through radiation from exposed surfaces.

By referring to fig. 127, it will be seen that rarefication of the condenser is effected by steam jet (2) followed by the action of kinetic ejector (4) supplied with water by the kinetic pump through pipe (5), and discharging into the kinetic tank (13).

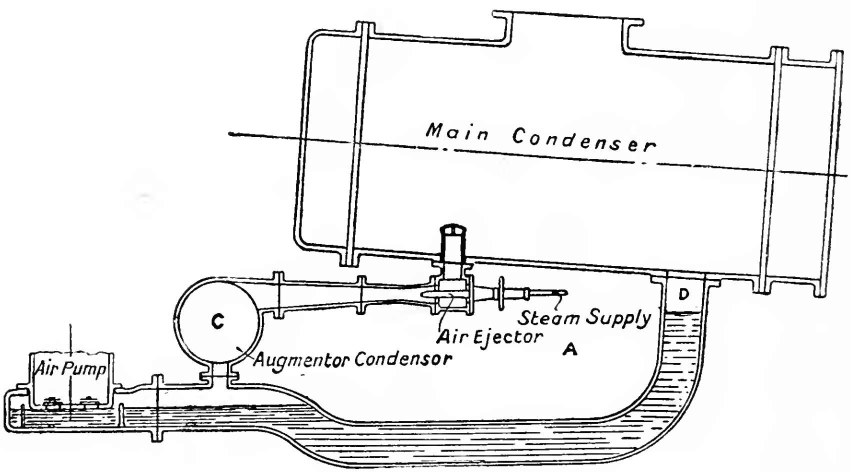


Fig 127*a*.

Water of condensation flows through pipe (7) into the head pump, whence through stand pipe (8) to the pressure pump, and thence through pipe (9) and non-return valve (10) into the kinetic tank. The excess of water in the tank corresponding to the feed-water is delivered by the kinetic pump through pipe (11) and float controlled valve (12) into the feed tank, which may be placed overhead.

Several Types of Rotary Air-pumps have been introduced to steam users with varying success from time to time, the majority of such pumps making use of the action of centrifugal force on water particles moving in small channels in the rotating part of the pump, or projected at a high velocity in thin films across the air suction chamber, the air being removed by becoming imprisoned between these moving particles or films. The disadvantages of many examples of this type of pump are said to be that:—

The tendency to rapid clogging and choking of the small passages with dirt or oil, necessitating the frequent dismantling of the pumps for cleaning.

Extreme sensitiveness of the pumps to sudden changes of pressure in the condenser.

Great sensitiveness to changes of the water temperature.

Inability to deal satisfactorily with any serious increase of air leakage above the normal figure.

Low thermal efficiency, all frictional expenditure of energy in the moving parts being generally lost to the system.

Parsons' Vacuum Augmentor was the means devised by Sir C. Parsons to obtain the high vacua so conducive to the high efficiency of the turbine, and so necessary when a L.P. turbine is added to a reciprocating engine to increase the economy of power generation.

The vacuum for brake purposes on a railway train is produced by the same means as here provided for getting rid of the gaseous products of a condenser; and Mr. Rodger thirty years ago introduced the "vacuum blower" for the purpose of forming a vacuum in the condenser of a compound engine, to ensure the easy handling of that engine. The adaptation of these means for exhausting a condenser was made by Parsons with success, and he is able to obtain quickly, and maintain steadily, quite a high vacuum, and higher than possible for the same air-pump to attain without it.

By reference to fig. 127*a* it will be seen that the air-pump is placed well below the condenser, so that there is a column of water between them, which acts as an air seal. The air ejector A draws from the condenser bottom free from condensed water, and discharges into a small auxiliary condenser C, whose surface is only about 5 per cent. that of the main condenser. In this condenser the gaseous products are cooled and freed from water vapour which is thrown down and drains away to the air-pump, to which also the cooled gases are projected. The difference in level of the pump bottom and condenser bottom, 3 to 4 feet, causes a difference of $1\frac{1}{2}$ to 2 inches of vacuum. The ejector being always in operation, the condenser is kept constantly free from gaseous matter, which interfere so seriously with the efficiency of the cooling surface when present with water vapour. By such means, not only is the vacuum maintained high, but the cooling water required is the minimum quantity when the efficiency of the surface is a maximum.

Pump Rods.—The vertical single-acting pump in oscillating paddle-wheel engines is generally worked by a connecting-rod from a crank, or by means of a large eccentric, whose rod passes through a trunk cast with the bucket, and connected to a socket fitted to the bucket, and secured by a brass cap-nut underneath, or by means of a lever, as in fig. 20.

$$\text{Area of section of the rod} = 0{\cdot}01 \times \text{area of pump bucket.}$$

Or in case of a round rod

$$\text{The diameter of rod} = 0{\cdot}1 \times \text{diameter of pump.}$$

When two bolts are fitted to connect the brasses, etc., at the end,

$$\text{The diameter of each bolt in the body} = {\cdot}056 \times \text{diameter of pump.}$$

When the air-pump is of the single-acting type in screw engines, it is generally worked by means of a rod, either of one of the zinc bronzes rolled or iron cased with brass,* having a tapered end fitting into the piston or bucket, and secured by a nut on the top side, or tapered the other way and secured by a cap-nut of brass (fig. 116) on the bottom side.

The diameter of the rod = ·15 × diameter of pump.

It is no unfrequent thing, though, in vertical engines, to find the pump worked by a connecting-rod and trunk as in paddle-wheel engines, the motion being derived from levers worked from the piston-rod crosshead in the same way as for pumps with fixed rods.

The arrangement of the pumps is various as regards detail, but for vertical screw engines is generally alike in principle, and may be divided into three systems—(1) Air-pump and circulating pump, both single-acting, and each worked separately by levers, etc., from a piston-rod crosshead; or air-pump and circulating pump side by side (the latter being either single or double-acting), and worked by levers from one piston-rod crosshead; or a single-acting plunger circulating pump inverted over the air-pump, and having a common rod, crosshead, levers, etc. (2) Air-pump and circulating pump worked each from a piston direct, or from the same piston-rod crosshead. (3) Both pumps worked by eccentrics on the crank-shaft, or by a crank-pin on the forward end of the crank-shaft. (4) The air-pump worked in one of the ways above described, and the circulating pump a centrifugal driven by an independent engine. (5) The air-pump detached and worked by an independent cylinder or engine, and the circulating pump centrifugal and worked by another independent engine. (6) The air-pump and circulating pump combined and worked by the same independent engine, the latter being centrifugal direct and the former by gearing.

The first and fourth systems are the ones most generally adopted, the sixth is specially suitable with turbines.

When the two pumps are worked independently of each other, each from a crosshead by levers, the strain from the pumps is distributed, and their weight tends to balance the pistons; but *per contra* the expense of working parts is increased, and the number of parts requiring attention and liable to accident is doubled; more space is required, and, as the low-pressure piston requires the greater counterbalance, it is an advantage to have both pumps worked from it.

If the two pumps are side by side, their rods are connected to a common crosshead, etc., thus leaving the space on the forward or after side of them free for the condenser, and admitting of more elasticity in its design; and, as is generally the case, the pumps are worked from the after cylinder, the condenser is easy of access, and there is plenty of space for the tubes to be drawn in a fore and aft direction.

The circulating pump inverted over the air-pump is a very convenient arrangement for small engines; but as the strains from the circulating pump have to be taken by the stand pipes from the pumps and the condenser top, it is found to give trouble in large engines; also, when the ship is light, and the valves out of order, it fails to pump well; but a large amount of space is saved by this plan, and it was at one time very frequently found in small ships.

The piston-buckets are made of bronze (in one thickness generally) stiffened by from 4 to 8 ribs; the circumference is either recessed down,

* Non-corrodible steel can now be obtained quite suitable for pump rods.

so as to admit of gasket being coiled around it, or else it is formed like a stuffing-box, and fitted with a gland or junk-ring, secured by studs and nuts, which pass through lugs cast with it, so as to jam the packing tight after it has been placed in.

Air-pumps are sometimes fitted with spring rings and junk-rings, similar to those of a steam-piston, but made of gun-metal instead of cast iron; this, however, has been but rarely done, and then only in very large pumps, as in ordinary cases hemp packing serves the purpose very well, and is much cheaper. Vertical pumps will work sufficiently well without packing; but, when required, bronze Ramsbottom rings answer the purpose quite well.

The diameter of the horizontal air-pump rod = $\cdot 15 \times$ diameter of the pump + $\frac{1}{4}$ inch.

The rods working in this pump are of hard rolled bronze or Muntz metal; the Admiralty now prefer the hard rolled bronze, and have, therefore, struck the latter out of their specifications as being too soft for continued wear. When the rods are of large size, they are sometimes made of wrought iron, cased with gun-metal, but unless the rods are large and long, it does not pay to case them, except for the sake of the harder surface (v. footnote, p. 390).

The rod fitting into the steam-piston is, of course, of wrought steel, and is connected to the brass rod by a box-coupling and cotters, the socket being formed with the steel rod, in order that the piston-bucket may be drawn out from behind for examination without removing the rod.

The pump-rod is usually fitted into the bucket in a similar manner to that of a piston-rod, the taper at the end being about 1 in 12.

Pump Buckets.—The following rules give the dimensions of an ordinary pump bucket, of which fig. 128 is an example :—

$$x = 0{\cdot}3 \times \sqrt[4]{D} + 0{\cdot}15 \text{ inch.}$$

D is the diameter of the pump in inches.

The thickness of the disc when solid	$= 1{\cdot}0 \times x$.
,, ,, ,, perforated	$= 1{\cdot}7 \times x$.
,, ,, flanges at edge	$= 1{\cdot}1 \times x$.
,, ,, metal around rod end	$= 1{\cdot}5 \times x$.
,, ,, ,, the rim	$= 1{\cdot}0 \times x$.
,, ,, packing	$= 1{\cdot}1 \times x$.
,, ,, ribs	$= 0{\cdot}8 \times x$.
The breadth of the packing	$= 4{\cdot}0 \times x$.
,, depth of bucket at the middle	$= 6{\cdot}0 \times x$.
,, number of ribs, one for each 4 inches of diameter.	

The bronze liner is usually from $\frac{1}{2}$ inch to $\frac{7}{8}$ inch thick, or its thickness $= 1{\cdot}1\,x - 0{\cdot}2$ inch.

This liner is let into the cast-iron bottom, and made to fit tight at both ends, facings on both liner and bottom being formed for that purpose; these facings are made with a very slight taper, so as to allow of a little *draw* on fitting in; the liner is then secured by brass screws passing through and through. The pump-barrels of vertical pumps are usually of bronze, $\frac{5}{8}$ to 1 inch thick, secured to the foundation or pump bottom by flanges cast with them; the *head* valve-boxes, etc., are also borne on them, and secured by flanges in a similar manner (v. fig. 122).

Pump Valves.—The arrangement of the valves of an air-pump is of the highest importance, for, as has been stated, the efficiency of the pump to a very large degree depends on it; especially is this the case with modern compound engines, where but a small amount of water is pumped at each stroke. A triple-compound engine indicating 2,000 horse-power uses about 8 cubic feet of water per minute, so that for a single-acting pump, with 100 revolutions of the engine, the amount per stroke is only 138 cubic inches; in practice, however, the pump fails frequently to take this amount, and so, until it accumulates behind the foot valves, no water enters the pump. Again, a double-acting pump is seldom so efficient at the front end as it is at the back, and, consequently, until there is a glut of water, the front ceases to pump. From causes of this kind trouble was often experienced with the double-acting pump of surface-condensers, and a remedy that has answered very well indeed in obviating this is to connect the space between the valves to the hot-well by a pipe about ·15 the diameter of the pump, having a suction or non-return valve fitted to it, so that the pump is to some extent continuously charged with water.

There are certain conditions that should be most carefully observed in arranging the valves, and they are—

(1) The foot valve should be in such a position that the water readily drains by gravity from the condenser into the pump.

(2) The valves should be designed so as to open with the least possible pressure under them, and still readily close on the return stroke of the pump.

(3) The flow from the condenser to the hot-well should be as direct as possible—that is, there should be as little change as possible in the direction—and no obstacles, so that the velocity generated at leaving the condenser should not be checked until the water is in the hot-well.

(4) The space between the head and foot valves of a double-acting and the head and bucket valves of a vertical single-acting pump should be a minimum, and all recesses, etc., where air can lodge or eddies can form, should be avoided without fail.

To drain the condenser efficiently, the foot valve should be below its lowest level, and, when possible, the channel from it to the pump inclined so that the water naturally flows in that direction; all stiffening ribs, etc., should be placed on the outside, so as to avoid gutters in which the water can lie.

The foot valves can only open by the pressure under them being greater than that on them, and when, as in the case of surface-condensers, the head of water is not sufficient to do so, it depends entirely on the pump forming a better vacuum between the foot and head valves than there is in the condenser. Now, if there be a good vacuum in the condenser, it requires the full efficiency of the pump to produce a better one between the foot and head valves. At most there can be but little difference between the pressure in the condenser and that in the pump-chamber, and if the valves have much resistance in themselves they will cease to act. The full force of this is better appreciated when it is remembered that the pressure in the pump-chamber must vary from slightly above atmospheric pressure, when the head valves are opened, to below that in the condenser in order to open the foot valves. For this reason, if the space between the head and foot valves is more than one-fifteenth the capacity of the pump, it is impossible to obtain a good vacuum in the condenser. This clearance space is virtually reduced by filling it with water; without such virtual reduction very few horizontal air-pumps would be capable of maintaining over 20 to 22 inches of vacuum in the condenser.

The makers of the horizontal air-pump usually arrange the foot valves so that they are either on a vertical plane seating or on one inclined so as to allow of the valves opening easily. A brass box, having valves on five of its sides, and bolted to the condenser by the sixth, so as to cover the outlet orifice from the condenser to the pump, was sometimes used. In this way the clearance space is partly filled, and five valves can work in the space usually occupied by one; three of the valves are in vertical planes and open horizontally, the two others are in horizontal planes, one opening upward and one downward.

The head valves should also be carefully arranged, for although not so difficult to place as are the foot valves, a little want of forethought may cause a badly working pump. They should always be so arranged that water lodges on them, so that in case they leak only water, *and not air*, escapes back into the pump; the valve seats and parts adjacent should be free from pockets underneath in which air can collect, and for this purpose they should be in the highest part of the pump-chamber, and if any part is partitioned off by a stiffening web or fillet, a communication should be made from its top to the underside of the head valves.

Since mineral oil has been used as the lubricant for cylinders, ordinary india-rubber ceased to be used for air-pump valves, inasmuch as that oil is a solvent of the rubber. Many attempts have been made to manufacture a rubber which will withstand the oil, but none of them have been perfectly successful, and only a few have succeeded in preventing the rapid action noticeable on nearly pure rubber. Many engines are now kept going with no supply of oil internally beyond what enters with the piston-rods.

Vulcanite Valves.—India-rubber can, by a certain process, be converted into a hard black substance resembling ebony, and hence sometimes called *ebonite;* this substance is light and very strong, and quite impervious to oil, and has been used with great success for air-pump valves. The valves made of vulcanite (fig. 128*a*) are generally circular and flat, strung on a stud through the middle, on which is a flat brass guard of the same diameter as the valve, and against which the valve lifts bodily. The wear of these valves is very little indeed. Valves of fibre are also fitted in this way.

Metallic Valves.—In the old days the air-pump valves were usually of cast brass, and of large size, the foot and delivery valves being hinged or "flap" valves, and the bucket valve annular. The flap valve is still employed in cargo ships where the foot valve is not accessible or easily got at, and answers the purpose very well indeed; it should be of ample size, and have very little lift.

Coe & Kinghorn's Patent (fig. 129) consists of tongues made of very thin rolled sheet phosphor bronze. These tongues or flaps cover a grating in the same way as india-rubber, and are fitted with curved guards to admit of a gradual bend. These valves work very well, but great care is necessary in making and setting the guard, so that when the valve is open there is no change of flexure or angle on which the flap can work and gradually break.

Thompson's Patent (fig. 130).—A large number of small saucer-shaped valves, of phosphor or manganese bronze, cast very thin indeed, are sometimes fitted, and, being very strong and comparatively light, do the work very well. Single discs of thin sheet-brass or phosphor bronze have also

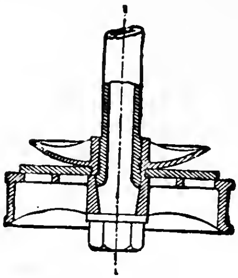


Fig 128.

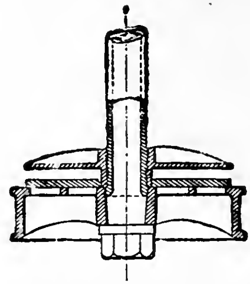


Fig. 128*a*.

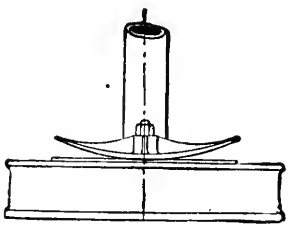


Fig. 129.

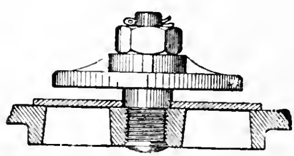


Fig. 131 —Flat Disc Valve.

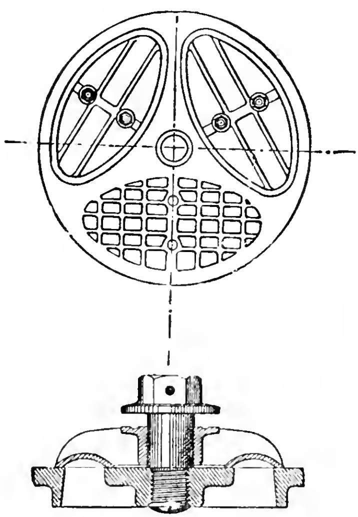


Fig. 130 —Annular Valve.

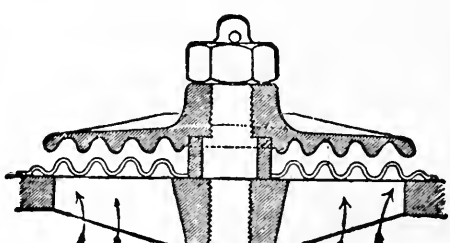


Fig. 132.

Figs. 128-132.—Air-pump Buckets, Valves, etc.

been used for this purpose with success. Thin discs of brass are often used; they are inexpensive and work very well. Discs of fibre also answer very well.

The valve shown in fig. 131 is simply a thin, metallic disc without stiffening of any kind. When very little lift is given to this valve it works very well, and rarely gets out of shape sufficiently to be seriously inefficient. It is sometimes fortified by a second and a third thin disc, each one smaller than the one on which it is superposed, and having holes through it on a circle of less diameter than those in the disc below it. In this way the lower disc permits water to flow around its edge and through the holes in it, and when closed the holes in it are covered by the next disc above it.

Beldam's Patent Valve (fig. 132).—In this case the valves are made of very thin sheet metal and stiffened by a series of circular corrugations. This valve has answered the purpose very well, for, while mantaining an even flat surface to cover the sea, it is elastic and can give under an unfair pressure and return to its normal form without permanent distortion.

Area through Valve Seats.—The area of opening past the valves depends on the size and velocity of the pump, and should not be less than will admit the full quantity of water for jet-condensation at a velocity not exceeding 400 feet per minute. In actual practice the area was generally in excess of this. If the foot valves are large they will be sluggish in action; if they are small the velocity of water, etc., will be sufficiently high to raise the valves and keep them open by the energy of the particles striking them; this argument especially applies to the pumps of surface condensers. It may be noted that the vertical pumps of the old jet-condensing engines were frequently without foot valves.

Suction Pipes from the Condenser to the Air-pump of marine engines have in the past been made often unnecessarily large. Experiments have shown that very little reduction in vacuum is made when they have been throttled down to quite small areas of cross-section, in one case to one-tenth of the original without any reduction. If D is the diameter of the air-pump in inches, and S the stroke in feet, and R the number of revolutions—that is, the number of lifts per minute—then

$$\text{Diameter of suction pipe in inches} = \frac{D}{48}\sqrt{S \times R}.$$

$$\text{Diameter of delivery pipe in inches} = \frac{D}{33}\sqrt{S \times R}.$$

If there were no fear of a glut of water coming over at any time from the air-pump its delivery pipe might be much smaller, but there is always that danger from the tubes splitting, corroding, or drawing out of place, or in a seaway with a list there might easily be such a glut from many condensers. The mall suction pipe is, therefore, a means of safety at all times.

$$\text{The area through the foot valves when fitted} = \frac{D^2 \times Z}{1{,}300} \text{ in square inches.}$$

$$\text{Area through the bucket valves} \quad ,, \quad = \frac{D^2 \times Z}{1{,}170} \quad ,,$$

$$\text{Area through the head valves} = \frac{D^2 \times Z}{1.050} \quad ,,$$

Z is the speed of pump in feet per minute $= 2\,S \times R$.

An air-pipe should always be fitted to the hot-well, as high up as possible, and its diameter should be $\frac{1}{3}$ that given above.

The bucket of a single-acting pump must be sufficiently large to admit of a valve area through it not less than given above. To obtain this, they are better designed with a short stroke, although the perimeter and the source of leakage is thereby greater. It is usual now and certainly advisable to deliver the water from the air-pump into a tank about 12 times the capacity of the pump; the delivery pipe is in diameter 0·7 that given above. The air-pipe of the tank should be about half the area in section. The tank should have an overflow pipe and a glass gauge. The suction for the feed-pumps should be near the bottom and fitted with a strainer.

Circulating Pumps.—Two kinds of pumps are employed to circulate the cooling water in the condenser of the mercantile marine—the ordinary single- or double-acting reciprocating pump, and the rotary pump. The single-acting pump is usually fitted to small engines, and the double-acting to larger ones. The latter is preferable to the former, but more expensive. They are generally worked by the main engines, but sometimes by independent ones.

Single-acting Pump.—This is used in vertical engines, and is similar to the single-acting air-pump already described. It works very well, and when provided with an efficient air-vessel, the flow is fairly steady. In very small engines a plunger-pump is used, and formerly inverted plunger-pumps were fitted to engines of considerable power. A good pet-valve, which will admit air to the pump, but not allow the water to pass out, should be fitted; and likewise a pass-valve, which opens a communication between the delivery and suction, is a requisite; the former prevents noise by providing an air cushion for the water, and the latter checks the supply without straining the pump.

Double-acting Pumps.—These give a steadier flow of water, and cause less shock and strain on all the working parts, pipes, etc., than do the single-acting pumps; but even these should be fitted with pet-valves and pass-cocks or valves.

Size of Circulating Pump.—The capacity of this pump depends on the quantity of cooling water and the number of strokes per minute.

Let Q be the quantity of cooling water in cubic feet, and n the number of strokes per minute, and S the length of stroke in feet.

$$\text{Capacity of circulating pump} = \frac{Q}{n} \text{ cubic feet.}$$

$$\text{Diameter} \quad ,, \quad ,, \quad = 13{\cdot}55 \sqrt{\frac{Q}{n \times S}} \text{ inches.}$$

Example.—To find the diameter of a double-acting circulating pump of an engine condensing 2 cubic feet of water per minute, and requiring 40 times the amount of cooling water; the stroke of the pump is 18 inches, and the number of revolutions 120 per minute.

Here $Q = 40 \times 2$ or 80 cubic feet; $n = 120 \times 2$ or 240.

$$\text{Diameter of pump} = 13{\cdot}55 \sqrt{\frac{80}{240 \times 1{\cdot}5}} = 6{\cdot}4 \text{ inches.}$$

The size of the circulating pump is to a large extent dependent on the same conditions that determine the size of the air-pump, and may, therefore, bear a constant relation to the size of the air-pump; and since the size of the air-pump is often determined by the size of the cylinders, that of the circulating pump may be found in a similar manner. When the circulating pump is *single-acting*, the capacity should be 0·038 of that of the low-pressure cylinder, and when the circulating pump is *double-acting*, 0·021.

TABLE XLV.

Description of Pump.	Description of Engine.	Ratio.
Single-acting,	Expansive 1½ to 2 times,	13 to 16
,, ,,	,, 3 to 5 ,,	20 to 25
,, ,,	Compound, triple, and quadruple,	25 to 30
Double- ,,	Expansive 1½ to 2 times,	25 to 30
,, ,,	,, 3 to 5 ,,	36 to 46
,, ,,	Compound, triple, and quadruple,	46 to 56

For high vacuum in the tropics the sizes should be 50 per cent. larger.

When the air-pump is *double-acting*, the capacity of the *double-acting* circulating pump should be 0·52 of that of the air-pump, the double-acting circulating pump being more efficient than the double-acting air-pump.

Table xlv. gives the ratio of capacity of cylinder or cylinders to that of the circulating pump.

Circulating Pump-Rods are made of the same materials, and in the same way as for the air-pump, and when possible the rods of both pumps are made identically alike, so that one spare rod serves for both.

Diameter of circulating pump-rod = 0·22 × diameter of pump.

When the pump is double-acting, or of comparatively long stroke, = 0·22 × diameter of pump + ¼ inch.

Circulating Pump Bucket.—When single-acting it is similar to that of the air-pump, and when double-acting it is simply a piston of brass.

Although it is usual to pack these pumps with hemp gasket or bronze rings, there is no necessity for this, since the water flows freely into the pump by gravity, and the pump moves too quickly to allow of much leakage past the piston. Many engineers now dispense with packing, and simply make the piston a fairly good fit in the barrel: while others turn either a spiral groove, or a series of parallel grooves, on the edge of the piston, which has the effect of keeping the surface well lubricated and preventing leakage *when working*. The friction of the unpacked pump is considerably less than that of a packed one, and in fast-running engines this is no slight consideration. It may be taken now as certain that it is better not to pack the circulating pump of a fast-working screw engine, and packing is of doubtful advantage when the engine is slow-working, provided the pump is below the water-line.

Circulating Pump Valves.—These are almost always of the best india-rubber, and of the quality known as "floating," from the fact of the specific gravity of rubber, with only such slight admixture of foreign matter as to render it usable, being less than that of water.

Valve Area.—The clear area through the valve seats and past the valves should be such that the mean velocity of flow does not exceed 450 feet per minute. It is not always easy to obtain so large an area, but when the velocity of flow is high the valves wear out more quickly, and the resistance of the pump is considerable, so that every effort should be made in this direction.

The pipes should be of such a size that the mean velocity of flow* through them does not exceed 500 feet per minute when comparatively small, and long for their size; when large, and having fairly easy leads, the allowance may be as much as 600 feet. The suction pipe of a double-acting pump should be of the same size as the delivery pipe, but it may be considerably smaller when the pump is single-acting and below the water-level.

Diameter of Suction Pipes for circulating pumps may be obtained as follows, when the cooling water per I.H.P. is not more than 8 lbs. per minute, which is sufficient for 26 inches in the tropics and 28 inches in temperate zones, and the velocity of flow as above :—

$$\text{Diameter in inches} = \sqrt{\frac{\text{I.H.P.}}{25}} + \tfrac{1}{2}\text{ inch.}$$

This may be used for centrifugal pumps, as also for good double-acting pumps.

Let A be the area of the pump, D its diameter in inches, S the mean speed of movement in feet per minute; then

$$\text{Area past the valves is not less than } \frac{A \times S}{450}.$$

$$\text{Diameter of pipes} = \frac{D}{F}\sqrt{S}.$$

Suction pipe of small double-acting pump,				$F = 22.$
Delivery	,,	,,	,,	$F = 23.$
Suction	,,	large	,,	$F = 24.$
Delivery	,,	,,	,,	$F = 25.$
Suction	,,	small single-acting	,,	$F = 26.$
Delivery	,,	,,	,,	$F = 22.$
Suction	,,	large	,,	$F = 27.$
Delivery	,,	,,	,,	$F = 24.$

Both kinds of pumps should have air-vessels, and the single-acting pump should have one twice the capacity of the pump when possible, and never less than one-and-a-half times its capacity. When the water is *pumped through* the tubes, the doors of the condenser may be made with pockets, which serve as air-vessels in forming a cushion. Air-pipes should be fitted to the highest points of the waterways, when the water is pumped into the condenser, to allow the air to escape, so that it may run full, and always allow water to be in contact with the tubes.

Rotary Pumps.—One or two forms of rotary pump have been tried for the purpose of circulating the cooling water, but only the centrifugal pump achieved perfect success.

* The velocity of flow should not exceed $350\sqrt[3]{\text{diameter of pipe}}$.

The advantages of the rotary pump are—

(1) There are no valves, etc., to interfere with the flow of water, or to get out of order.

(2) Being easily worked by an independent engine of small size, or a motor, it is usually so provided, and can be then started before the main engines are moved, and so keep the condenser cool during the process of "warming through" the cylinders, etc., and also while standing.

(3) Having this independent engine, the supply of cooling water is varied to suit the varying circumstances, and the power required to work the pump varies then with the quantity of water.

(4) The efficiency for low lifts is greater than that of a reciprocating pump.

(5) The supply of water is continuous, and enters the condenser without shock, thereby putting no stress on the castings, pipes, etc., beyond that due to the "head."

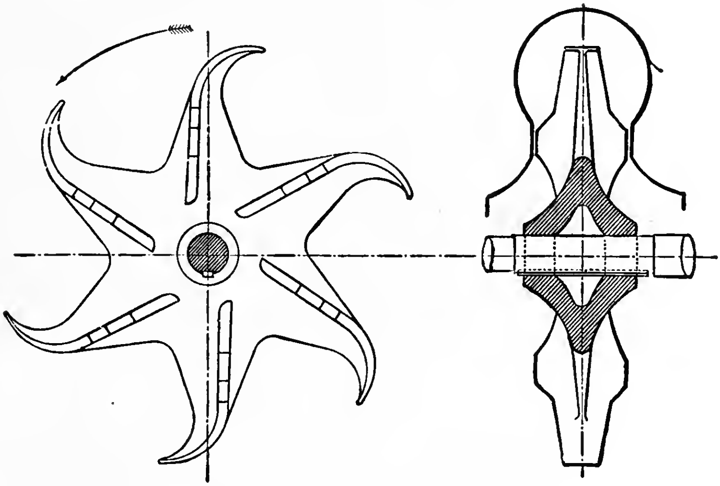


Fig. 133.—Wheel of a Centrifugal Pump.

(6) It is easily placed in the engine-room, and the absence of a reciprocating pump worked by the engine, especially in a horizontal engine, admits of a better design and arrangement of condenser and air-pump.

On the other hand, the centrifugal pump is somewhat more expensive than an ordinary pump, and requires some attention when at work; the latter objection is, however, now considerably lessened, although only by increasing the former.

Centrifugal Pump.—This essentially consists of a wheel (fig. 133) having thin vanes as arms, which act on the water so as to give it a circular motion in a cylindrical case enclosing the wheel; this case is provided with an enlarged chamber around it, into which the water from the wheel is whirled, and from which it escapes through a branch tangential to it (*v.* fig. 134).

The principle on which this pump works is, that a particle moving in a circular path is under the action of two forces, one tangential, and the other normal to its path, or at right angles to the tangential one; when this latter force ceases to act, the particle moves away in a path tangential to the circle at the point where the retaining force ceased to act. In the centrifugal pump, the particles of water flowing into the centre of the pump are gradually put into motion and whirled round until, after a spiral course, they

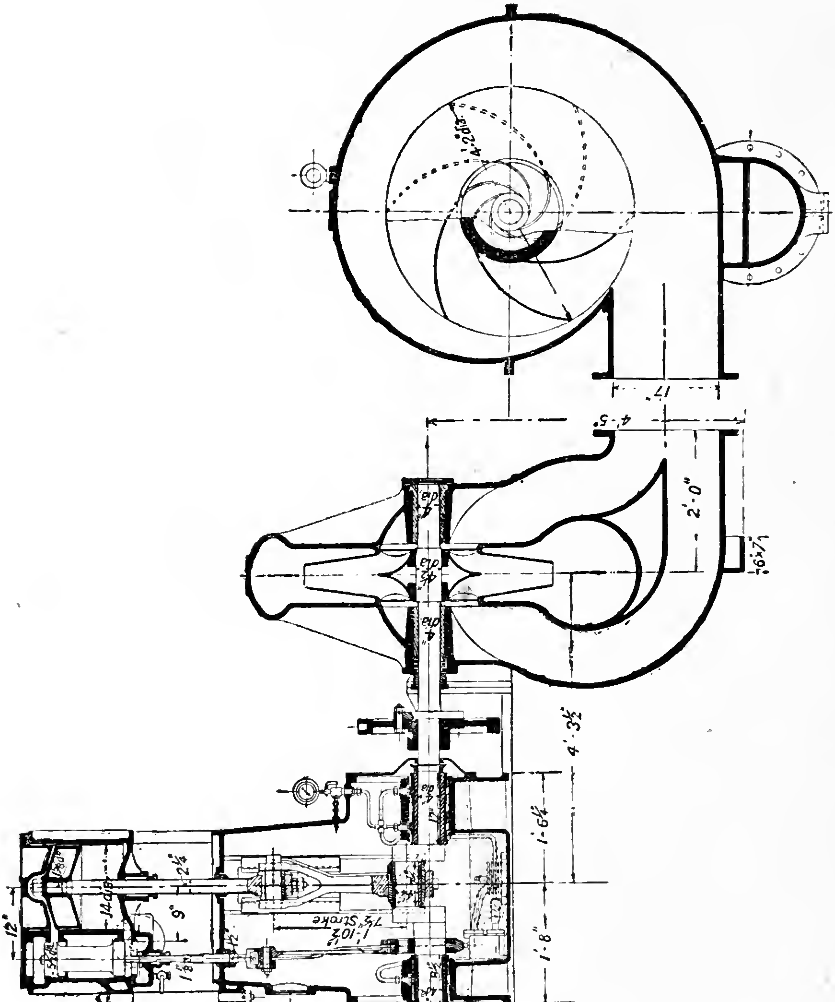


Fig. 134.—17-inch Circulating Pump. (By Thwaites Bros., Ltd.)

arrive in the outer channel, and there are retained in a circular course till they reach the outlet, where the retaining or normal force ceases, and they fly away tangentially through the outlet. The vanes of the wheel are sometimes enclosed between two discs of thin metal, to which they are attached, or they fit closely to the sides of the case in which they work.

Experience has shown that the best form of wheel is one whose vanes are sickle-shaped, as shown in figs. 133 and 134, and there is less resistance if the vanes fit the pump-case instead of having the enclosing discs, inasmuch as these latter have considerable friction on their surfaces, especially on that next the pump-case.

The outer passage, or *whirl chamber*, is usually formed like a snail, so that its sectional area gradually increases from nearly nothing to the full area of the discharge pipe.

The inlet pipe leads from the outer rim to the centre of the pump, having generally a passage on either side, so that the water is delivered on both sides of the wheel. For convenience these pumps are sometimes designed now with the inlet at one side only.

The diameter of the inlet and outlet pipes of a centrifugal pump for circulating purposes should be such that the velocity of flow does not exceed $350\sqrt[3]{\text{diameter in inches}}$, feet per minute. Hence, if W is the quantity of water in gallons per minute,

$$\text{Diameter of pipes in inches} = \sqrt{\frac{W}{12\sqrt[3]{D}}} \text{ or } \sqrt{\frac{W}{0{\cdot}0343 \text{ velocity}}}.$$

For triple and quadruple engines the following rule gives ample size :—

$$\text{Diameter of pipe in inches} = \sqrt{\frac{\text{I.H.P.}}{C}}.$$

C = 16 for all ships liable to be employed in the tropics, and requiring high vacuum, 28 inches.

C = 25 for ships employed only in temperate zones.

The diameter of the fan-wheel or impeller is usually from $2\frac{1}{4}$ to 3 times that of the pipes, and always is such that the velocity at its periphery at full speed is in excess of 400 feet per minute, and generally from 500 to 800 feet.

The blades are curved as shown, that the water on entry may be gradually set into circular motion, and caused to flow into the whirl chamber with as little force as possible, and to leave the impeller also gently.

The size of the cylinder for driving the fan can be calculated from the usual conditions ; in practice its diameter is generally about $2{\cdot}3\sqrt{\text{diameter of pipes}}$, and its stroke 0·25 the diameter of the impeller.* When the velocity of flow is high, as it often is nowadays for the purpose of keeping high vacuum in summer time, the cylinder must be larger even to the extent of 20 per cent. in diameter.

The impeller should be always of bronze, and made as thin as possible in the blades ; the spindle should be of strong bronze or incorrodible steel. The Admiralty require the spindle to be cast solid with the impeller. No doubt there is considerable danger of it working loose if not well fitted and secured to the spindle. This, however, can be done with care when a forged or other strong form of spindle can be used, whereas by the Admiralty method only a casting, with possible spongy places, is allowed.

As these pumps are sometimes required to pump out the bilges and ballast tanks, it is usual to fit some means of exhausting the wheel-case of

* Large pumps should have compound engines with a low-pressure cylinder of the size thus calculated ; the high-pressure cylinder should be half the diameter when the boiler pressure is 120 lbs. and upwards.

air, so as to cause the water to flow up into it, as the fan has no sucking power, and will not draw until fully charged with water. For this purpose some engineers fit a small rotary exhauster, worked by means of a belt from the spindle, while others fit a small steam-ejector on the same principle as Rodgers' vacuum blower. When no such special means is provided, the pump may be charged from the sea, and the sea inlet left slightly open until the pump begins to draw from the bilge or tanks. In the Navy the circulating pumps are arranged to draw from the bilges; for this purpose throttle-valves are fitted to the suctions, so that the sea supply may be instantly shut off and the bilge turned on, this method having been found the most successful for this purpose.

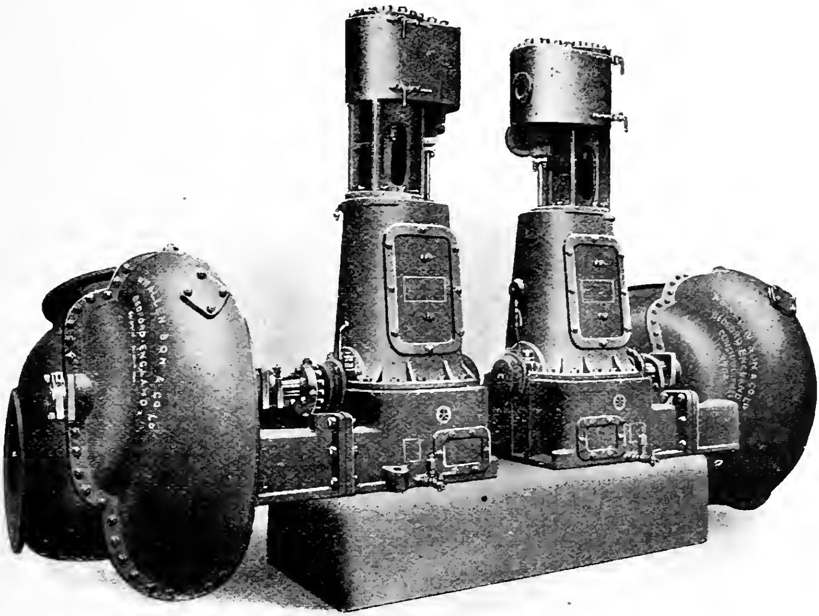
Centrifugal Circulating Pumps of R.M.S. "Mauritania" are each 32 inches in diameter of delivery pipes; there are a pair of such pumps to each set, and a set in each engine-room capable of delivering 45,000 gallons per minute, or 450.000 lbs. The total steam consumption at full power is about 1,000,000 per hour, or 16,667 per minute in that ship—that is, 8,334 lbs. is condensed in each condenser—there is, therefore, pump power for 54 times the quantity of water condensed. In modern naval ships the engine circulating pumps are capable to supplying 45 to 60 times the weight of steam produced for full power, cargo steamers 40 to 50.

Feed-pumps.—The duty of the feed-pump is to supply the boiler with sufficient water to meet its requirements. It is obtained from the hot-well, so that when there is a surface condenser it is practically pure water, and the amount, under ordinary circumstances, is the same as that evaporated in the boiler; but owing to leakage and waste by blowing the steam whistle, or using an auxiliary engine which exhausts into the air, the quantity of water condensed is not always sufficient to make up for that evaporated. It is found necessary also occasionally to blow some of the water out of the boiler, to get rid of scum floating on the surface of the water, and this waste must be made good by a supply from the sea or elsewhere. To prevent a deposit of salt, etc., in the boiler, it was customary to blow out some of the very dense water from the boiler at fixed intervals, and as the blow-off cock was situated near the bottom of the boiler a considerable amount of solid matter was thus got rid of; the quantity of water blown out was made up by an extra supply from the sea. Now, since the surface condenser might leak, the pumps of engines having surface condensers were formerly made sufficiently large when worked by the main engines to do duty under such circumstances.

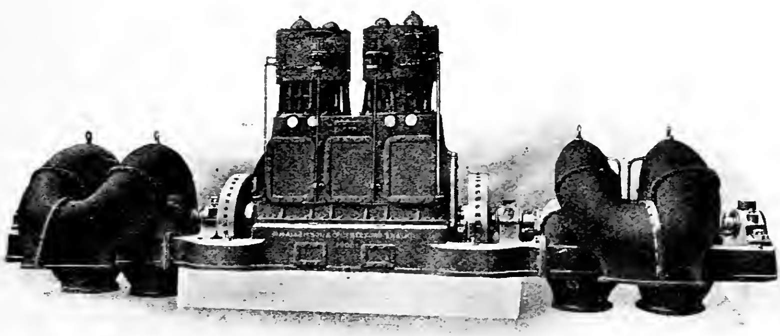
Sea-water contains about $\frac{1}{32}$ of its weight of solid matter* in solution; its density is measured by a hydrometer, which in the Navy is marked with degrees, so that when floating in pure water the zero point is at the surface, and when in clean sea-water it marks 10°. In the merchant service, engineers are accustomed to speak of the density by the number of ounces of solid matter to the gallon, sea-water containing 5 ounces to the gallon.

Net Feed-water.—The net feed-water is that quantity required to make up for what has been used as steam in the engine and its auxiliaries; let this be denoted by Q; let n be the number of times the saltness of the water

* On the British coasts the sea-water contains only about one fortieth of its weight of solid matter in solution.



(*a*) 1 Set for a Battleship 27,000 S.H.P. (W. H. Allen).



(*b*) 1 Set for R.M.S. "Mauritania," 65,000 S.H.P

Fig. 134*a*.—Centrifugal Circulating Pumps (Allen).

in the boiler is to that of sea-water, then

$$\text{The gross feed-water} = \frac{n}{n-1} \times Q.$$

This is the amount of feed-water which must be pumped into the boiler when salt water only is used, to maintain the saltness of n times that of the sea.

If the pumps were only of sufficient size to pump this amount of water, a considerable time would elapse before the boiler would be filled to the working level after "blowing off"; to meet this objection it was usual to make *each* feed-pump capable of pumping twice this quantity, therefore

$$\text{Quantity of water for each pump to supply} = \frac{2n}{n-1} \times Q.$$

Since a surface condenser supplies pure water for feeding the boiler, and the waste is now made up with fresh water, either carried in the tanks or distilled from sea-water, there is not the same need for blowing off the boilers, the feed-pumps may be, therefore, very much smaller than when jet condensation is practised, and are generally of such a size when worked by the main engine that *each* is capable of delivering three times the *net feed-water*, assuming them to have an efficiency of 1·0; the pumps, when both are working, can deliver six times the net feed, which is sufficient to satisfy any extraordinary demands.

If Q be the quantity of net feed-water per minute in cubic feet, l the length of stroke of feed-pump in feet, and n the number of strokes per minute,

$$\text{Diameter of each feed-pump plunger in inches} = \sqrt{\frac{550 \times Q}{n \times l}}.$$

If W be the net feed-water in pounds—

$$\text{Diameter of each feed-pump plunger in inches} = \sqrt{\frac{8{\cdot}9 \times W}{n \times l}}.$$

The following empirical formula will give such sizes as will be found in practice, and which will closely approximate to those given by the above rule :—

$$\text{Capacity of each feed-pump} = \frac{\text{capacity of L.P. cylinder}}{C}.$$

General Description.			Steam Consumption.			Feed.	Value of C.
Ship.	Service.	Engines.	Main.	Aux., &c.	Total.	Allowance	
			Lbs.	Lbs.	Lbs.	Lbs.	
Ordinary cargo, .	General.	Quadruple.	14·0	0·70	14·70	17·5	650
,, .	,,	Triples.	14·7	0·74	15·44	18·5	600
Passenger & cargo,	Long voyage.	Quadruple.	14·0	1·25	15·25	18·5	600
Express, . .	Short ,,	Triple.	15·5	1·50	17·00	20·0	550
,, Cold zone,	,, ,,	Turbines.	13·0	1·65	14·65	17·5	..
,, . .	Atlantic.	Turbines.	12·5	2·50	15·00	18·5	..

Net feed-water in pounds per stroke approximately = area of piston in inches × stroke in feet × absolute pressure at release ÷ 53,600.

Example.—To find the net feed-water in pounds for an engine whose cylinder is 50 inches diameter, and length of stroke 3 feet, the pressure at release being 12 lbs., and the revolutions 100 per minute.

$$\text{Net feed-water } \textit{per minute} = 200 \times \frac{1{,}963 \times 3 \times 12}{53{,}600} = 263 \text{ lbs.}$$

All engines over 80 N.H.P. should have two feed-pumps, each capable of supplying the boilers when the engines are at full speed; and each pump should be so arranged that it may be worked quite independently of the other, and easily put out of action when not required. This latter condition is not always complied with in practice; but if it were so there would be then practically a spare pump, and only one in danger of derangement when the engine is at work.

The feed-pumps of vertical engines are usually moved by the same parts as operate the air and circulating pumps. They are sometimes fixed to the crosshead of these latter pumps, and sometimes driven from studs in the sides of the rocking levers. When there is only one set of rocking levers this latter plan should be adopted; for by placing one pump on each side of the lever centre they may both be used together and deliver alternately, and give a steady flow of water. In the Navy the use of feed-pumps worked by the main engines is entirely discontinued, and in the mercantile marine it is fast becoming the practice, especially in large ships, to feed the boilers by an independent pump or pumps, which are generally self-regulating. In fact, with all fast-running engines the feed-pumps are better detached from them and worked with absolutely independent automatically-controlled engines, so that the supply of water is steady and not intermittent, and exactly in proportion to the demand. The usual plans for treating the feed-water are the same in principle, though differing in detail. The water is delivered from the hot-well to a receiving tank by the air-pump in the mercantile marine, by a special pump in the Navy. The feed-pumps, two in number and in duplicate, draw from this tank at a uniform rate under normal circumstances, but if the supply in the tank fails the pumps are slowed down and stopped by a float arrangement connected to their stop valves. If the feed is wholly or partially shut off in the boiler-room the extra load that then comes on the pumps slows them down, and eventually stops them if "feed" is wholly shut off. As a rule, only one of these pumps works at a time, and is capable of supplying two to three times the *net feed* when working full speed (*v.* fig. 186).

Relief Valves should be fitted to each pump when worked by the main engine, and they are so arranged that each may work separately; when they both deliver into a common pipe, one relief valve is sufficient; these valves should be loaded to $1\frac{1}{2}$ times the boiler pressure.

Valves and Valve-boxes should always be of bronze; and since the seats as well as the valves wear out rapidly, they should be made separately from the box casting of harder bronze. The valves should have a seating area equal at least to 20 per cent. of the valve, and it is better to make them flat rather than conical. Some engineers use bronze balls fitting into conical seats; since they are constantly changing their position on the seats these balls wear very slightly and keep very tight. When the boiler pressure did not exceed 30 lbs., the valves of the feed-pumps were usually made of india-rubber; and some American engineers employed this material notwithstanding the increased pressure. If the rubber is thick and capable of withstanding the action of mineral oil, no doubt it will work well; but so much reliance cannot be placed on it as on the metal, especially at high pressures.

Bronze valves (fig. 119) make a noise when working, owing to the quick return to their seats on the pump ceasing to deliver: attempts have been made to reduce the noise and wear in various ways. The noise of the valves may be stopped by loading them with light springs made of steel nickel-plated, or of hard brass. Several small valves with little lift is found to be more satisfactory than one large one; for example, to obtain sufficient area through the seats of six valves to take the place of one 4 inches in diameter, each would be $1\frac{5}{8}$ inches, giving the sum of the perimeters 30·6 inches against the 12·56 inches of the 4-inch valve. It follows, then, that if the single valve lifted half an inch, the small one need only lift one-fifth of an inch. Such small valves must be forged of hard, tough bronze, carefully formed so that there is no groove or nick to form a starting point for a crack, as the tails quickly break off by the "jar" on seats and stops.

Air-vessels.—Each pump should be furnished with an air-vessel which may serve the double purpose of providing a cushion, and intercepting and collecting the free air from the new feed-water, which, when fresh or sea-water is used, is the active agent in producing corrosion when admitted to the boiler. To increase its usefulness in the latter capacity, a fine grating should be fitted to the inlet orifice, which will "spray" the water, and so separate the air, and a relief valve should be fitted to the top of the vessel, loaded to a pressure slightly below that obtaining when the pump is delivering at its maximum rate, but arranged to close when the vessel is nearly full of *water*. Air-vessels, however, are practically useless with the high pressures of steam now employed, except as a means of getting rid of gases and air.

Feed-pipes.—The pipes leading to and from the feed-pumps should be such that the average working velocity of flow does not exceed 500 feet per minute, and small pumps should have larger pipes in proportion, so that the flow through them does not exceed 350 $\sqrt[3]{\text{diameter in inches}}$.

Since the amount of water actually flowing through the pipes is generally very considerably less than the pumps are capable of discharging, the velocity is seldom more than half the above allowances; but as the pumps do occasionally deliver their full amount, the pipes must be large enough for that purpose.

If d is the diameter of the feed-pump plunger, and s its mean velocity in feet per minute, then

$$\text{Diameter of feed-pipe} = \frac{d}{28}\sqrt{s} + 0·25 \text{ inch.}$$

or, taking I.H.P. as the basis,

$$\text{Diameter of feed-pipe} = \frac{\sqrt{\text{I.H.P.}}}{18} + 0·25 \text{ inch.}$$

Example.—To find the diameter of the feed-pipes for a pump whose diameter is 6 inches, and the length of stroke 2 feet, worked from the levers of an engine, making 60 revolutions per minute. Here $s = 2 \times 60 \times 2$, or 240 feet.

$$\text{Diameter of pipe} = \frac{6}{23}\sqrt{240}, \text{ or 4 inches.}$$

Example.—To find the diameter of feed-pipe for an engine of 3,600 I.H.P.

$$\text{Diameter of pipe} = \frac{\sqrt{3,600}}{18} + 0{\cdot}25 \text{ inch} = 3{\cdot}58 \text{ inches.}$$

If there are two pumps which deliver alternatively, the pipes will be the same size throughout; but if the two pumps *may* deliver at the same time, the pipe beyond the junction of the two from the pumps must be nearly double the sectional area of one. As the resistance of pipes is due greatly to friction at the surface, and will consequently vary as the diameter, while the area of section varies as the square of the diameter, the resistance in the single pipe will be considerably less than the combined resistance in the two, and for this reason the sectional area may be less. In practice this area may be 0·8 of the combined area of the two. Hence, when there are two pumps delivering together—

$$\text{Diameter of main pipe} = 1{\cdot}265 \times \text{diameter of each of the branches.}$$

If there were two pumps, as in the first example, delivering together, the diameter of the main pipe would be $4 \times 1{\cdot}265$, or 5·06 inches.

"**Pet Valves.**"—Although it is prejudicial to admit air to the feed-water, it is sometimes necessary for the good working of all pumps to allow a little air to enter between the valves to form a cushion: they are, however, of no use to a modern feed-pump.

Feed Tank.—To avoid any waste of water through the overflow or air pipe of the hot-well when the feed-pumps are temporarily stopped, it is well to provide a tank, into which the water is discharged from the hot-well, and from which the feed-pumps draw. Such an arrangement is very beneficial in all engines, and especially in those having small hot-wells.

Feed-pump Rod.—This is of iron, and, in the case of hollow plunger pumps worked from a pin having a circular motion, it is jointed within the pump. As the plungers of vertical pumps are seldom without water in them, and difficult to empty, the joint should be such as will work with water as a lubricant. This is accomplished by bushing the joint with lignum vitæ, and casing the pins with brass; or white-metal (Fenton's) bushes, with steel pins, will do. The rod-end within the pump should be galvanised, or otherwise protected from being corroded.

When the rod is long and of iron, p being the boiler pressure.

$$\text{Diameter of feed-pump rod} = \frac{\text{diameter of plunger}}{40} \times \sqrt{p} + 0{\cdot}6.$$

If of brass, divide by 35 instead of 40.

When the rods are short, the diameter may be 0·7 of that given by the above rules.

The plungers, valve-boxes, and valves are always of best bronze, and the pump-barrel or case may be of bronze, but in the merchant service this is generally of cast iron. The same remark applies to the air-vessels, escape-valves, etc. The capacity of the air-vessel should be from 1·5 to 2 × the capacity of the pump.

Bilge Pumps.—These pumps, which are for the purpose of freeing the bilge of water, are somewhat similar in construction and method of working to the feed-pumps. Although the Board of Trade and Lloyds require that all steamships shall have two bilge pumps worked by the main engines, except in the case of engines under 70 N.H.P., they do not specify the capacity of these pumps. There is no definite basis of calculation for the size of these pumps, and it is generally at the caprice of individual engineers, many of whom still adhere to the old practice of making them of the same size as the feed-pump. They should not be, however, made less than given by the following rule:—

$$\text{Capacity of bilge pump} = \frac{\text{capacity of L.P. cylinder}}{350}.$$

The Capacity of the Bilge Pumps should really bear some reasonable relation to the displacement of the ship rather than to the size of the engines. The following rule, therefore, should be followed with this object in view:—

$$\text{Total bilge pump capacity in cubic inches} = 3{\cdot}5 \times D^{\frac{2}{3}},$$

D being the displacement in tons.

Example.—A ship of 5,000 tons' displacement should have bilge pumps whose capacity = $5{,}000^{\frac{2}{3}} \times 3{\cdot}5$, or 1,022 cubic inches. Such a ship, if a cargo steamer, might have a low-pressure cylinder 60 inches diameter × 42 inches stroke. Each pump would be by rule 340, and the two 680; on the other hand, the low-pressure of a faster ship might be 72 inches, and the two be then 1,000 cubic inches.

The plungers are usually of bronze, but in the merchant service are often of cast iron, and as this is harder, especially after the hard skin is formed by rubbing, they wear longer when of this material. With cast iron the neck and gland bushes should be of Fenton's metal to prevent corrosions setting it fast when not at work.

In the Navy the bilge pumps are now like the feed-pumps, quite detached from the main engines; in the mercantile marine, when the engines are run at a high number of revolutions, the same practice is followed; but when the engines are not quick running, as in cargo steamers, the bilge pumps are, as formerly, worked from the main engines. The Admiralty have at least one pump set apart for the bilges in each engine-room, and this is generally done in passenger steamers. There is also in each engine-room another pump which may be used for pumping out the bilges or for discharging water on deck; or one to do this service and the other to pump from the sea on deck or from the sea, and tanks to the boilers.

In the mercantile marine, the valve-boxes are of cast iron, and the valves often hinged "clacks," which are easy removed for cleaning. The covers of these boxes should be so made that they may be easily and quickly removed and replaced, and for that purpose are sometimes hinged, but better still, held down by two hinged bolts fitting into recesses in the cover.

The Board of Trade require that one engine bilge pump shall be arranged to draw water from the sea, and pump it on deck in case of fire. When this is done, the suction pipes should be fitted with a three-way cock, whose plug has only one port, so that it cannot be open to the sea and bilge at the same time, and so flood the ship. When the pipes are very large, it is not always convenient to fit such a cock, then a double valve-box with self-acting non-return valves is substituted; when possible, the cock should be fitted as the better and safer plan.

Directing Boxes.—It is required that the bilge pumps shall draw from each compartment of the ship. For this purpose, the suction from the pump is connected to the top of a box containing a series of valves by opening any one of which a communication is made to a separate compartment; the cover of each valve should have on it a label signifying to which compartment the valve opens a communication. These directing boxes should be placed in such a position that they are easily got at, and above the floor plating when possible, that the labels may be seen and the valves operated.

Mud Boxes.—Between the directing box and the pump should be fitted a box with a strainer, which shall intercept such solid matter as would derange the pump valves, if allowed to enter among them. They should have covers similar to those of the pump valve-boxes, and placed in such a position as to be easily got at for cleaning or examining.

Sanitary Pump.—In passenger steamers, it was usual to have a pump of about a half or one-third the capacity of the bilge pump, which could be put into gear, and worked by the engines, to discharge water on deck for sanitary purposes; this duty is now generally performed by a separate donkey pump.

Ejectors.—In the Navy, as in the merchant service, steam ejectors are often used for clearing the engine and boiler-room bilges. They are in principle very like the well-known feed injectors, without valves or other internal fittings liable to derangement from ashes, dirt, etc. They are started and stopped by merely opening and shutting a cock or valve, and require no attention when working, and are, consequently, most suitable for such a service.

The Power required to drive the Circulating Pumps may be estimated by taking the friction of the tubes and inlet and discharge pipes, allow for the loss of head due to entering and leaving the tubes, and assume the efficiency of the pump to be 0·6, and the engine driving it 0·65, so that the combined efficiency is 0·39. The "head" to overcome the friction in the tubes may be taken as $= \frac{L \times v^2}{2g \times d} 0{\cdot}60 = \frac{L \times v^2}{107\,d}$ feet, where L is the aggregate length of tubes in feet, d the diameter in inches, and g is 32 at sea level; v the velocity in feet per second.

For Example.—What head of water is necessary to overcome the frictional resistance of 30 feet of $\frac{3}{4}$-inch tubes, the velocity of flow being 8 feet per second. The internal diameter = 0·64 inch?

$$H = \frac{30 \times 64}{107 \times 0{\cdot}64} = 28 \text{ feet.}$$

The sum of the other resistances may be taken as a quarter of the above, so that the total "head" required to force water through a condenser with $\frac{3}{4}$-inch tubes so long would be 35 feet, or equivalent to a pressure of 15·2 lbs. per square inch.

TABLE XLVI.—Centrifugal Pumps for Circulating Water.

Main Engines.	Engine Served.		Boiler Pressure.	Diameter of		Steam Cylinder.		B.H.P. of Pump Engine.	Output 70° F.		Velocity of Flow.
	I.H.P.	Cooling Surface.		Discharge Pipe.	Impeller.	Diameter.	Stroke.		Per Minute.		
		Square Feet.	Lbs.	Inches.	Inches.	Inches.	Inches.		Cubic Feet.	Tons	Feet per Minute.
Naval triples, . . .	1,900	1,600	180	10·0	21	5·0	3·0	..	300	8·6	553
,, . . .	3,000	1,850	210	10·0	24	5·0	3·0	..	474	13·54	874
,, . . .	1,360	1,600	150	8·0	..	..	..	..	213	6·06	612
,, . . .	3,700	4,000	155	12·5	33	8·5	7·0	..	578	16·50	694
,, . . .	5,180	5,280	155	15·0	42	7·5	8·0	..	809	23·14	658
,, . . .	6,000	6,750	155	18·0	..	..	..	..	938	26·8	531
Mercantile triples, . . .	2,200	3,100	160	12·0	..	..	..	..	309	8·83	394
,, . . .	600	..	150	..	21	5·0	3·0	..	85	2·43	..
Mercantile turbine, . .	17,000	..	..	32	..	..	..	350	3,600	102·8	643
Standard, 30 feet head, . .	760 revs. per min. max.			7·0	22	5·0	4·0	16	152	4·24	562
,, ,, . .	695	,,	,,	8·0	27	5·0	4·0	22·2	208	5·80	600
,, ,, . .	640	,,	,,	9·0	28	6·0	5·0	27·0	272	7·59	610
,, ,, . .	590	,,	,,	10·0	28	6·0	5·0	28·7	288	8·04	520
,, ,, . .	510	,,	,,	12·0	30	7·0	6·0	34·0	368	10·27	470
,, ,, . .	470	,,	,,	14·0	32	8·0	7·0	47·0	510	14·28	467
,, ,, . .	400	,,	,,	16·0	39	9·0	8·0	61·0	640	17·86	457
,, ,, . .	390	,,	,,	19·0	42	10·0	9·0	92·0	960	26·80	487
,, ,, . .	360	,,	,,	22·0	48	12·0	10·0	130·0	1,360	38·0	504

The work done will be weight of water × 35 foot-lbs.

$$\text{The net horse-power} = \frac{W \times 35}{33{,}000}.$$

$$\text{The gross or I.H.P.} = \frac{W \times 35}{0{\cdot}39 \times 33{,}000} = \frac{W}{368}.$$

Example.—If the above condenser requires 12.000 lbs. of cooling water per minute.

$$\text{Then I.H.P.} = \frac{12{,}000}{368}, \text{ or } 32{\cdot}6.$$

TABLE XLVIA.—FLOW THROUGH PIPES NOT EXCEEDING 240 $\sqrt{\text{DIAMETER}}$ LONG UNDER ORDINARY WORKING CONDITIONS PER MINUTE.

Diameter of Pipe.	Water 60° F.		Live Steam. 200 lbs. Press. Abs.			Exhaust Steam. 3 lbs. Press. Abs.			Air Atmosphr. Press., 60° F.		
	Velocity.	Quantity.	Velocity.	Quantity.		Velocity.	Quantity.		Velocity.	Quantity.	
Ins.	Ft.	C. Ft.	Ft.	C. Ft.	Lbs.	Ft.	C. Ft.	Lbs.	Ft.	C. Ft.	Lbs.
1·0	350	1·909	5.000	27·19	11·71	3,300	17·95	0·152	1,000	5·438	0·418
1·5	401	4·917	5,535	69·51	30·00	3,653	45·94	0·389	1,107	13·90	1·070
2·0	441	9·600	5,945	129·7	54·12	3,923	75·60	0·646	1,189	25·94	1·995
2·5	475	16·19	6,285	213·9	92·12	4,148	141·2	1·196	1,257	42·78	3·290
3·0	505	24·80	6,575	322·6	138·9	4,340	212·9	1·804	1,315	64·52	4·963
3·5	531	35·45	6,835	456·3	196·6	4,511	301·2	2·552	1,367	91·26	7·020
4·0	556	47·95	7,070	616·8	265·6	4.666	407·7	3·456	1,414	121·3	9·331
5·0	598	99·00	7,475	1.019	438·9	4,933	672·5	5·700	1,495	203·8	15·68
6·0	636	126·7	7,825	1,536	661·5	5,165	1,014	8·590	1,565	307·2	23·63
7·0	670	179·2	8,130	2,180	939·0	5,362	1,419	12·02	1,626	436·0	33·54
8·0	700	244·3	8,410	2,938	1,263	5,550	1,939	16·43	1,682	587·6	45·20
9·0	728	321·7	8,660	3,826	1,649	5,715	2,525	21·40	1,732	765·3	58·87
10·0	754	431·6	8,895	4,852	2,100	5,867	3,202	27·14	1,778	970·4	76·65
12·0	800	628·0	9,305	7,300	3,144	6,116	4,818	40·83	1,861	1,460	112·3
14·0	844	903·6	9 670	10,357	4,012	6,382	6,836	57·91	1,934	2,071	159·3
16·0	882	1,229	10,000	13,950	6,000	6,600	9,207	78·00	2,000	2.790	214·6
18·0	917	1,550	10,295	18,018	7.760	6,795	12,000	101·7	2,059	3,646	280·5
20·0	960	2,096	10,570	23,050	9,930	6,976	15,220	129·0	2,114	4,611	354·7
25·0	1,023	3,485	..	..	..	7,378	25,210	213·7	2,236	7,626	586·6
30·0	1,087	5,345	..	..	..	7,725	38,000	321·8	2,341	11,033	848·7
35·0	1,145	7.620	..	..	..	8.030	53,600	453·6	2,433	16,480	1,268
40·0	1,197	10,470	..	..	..	8,290	71,800	608·5	2,514	22,500	1,731

CHAPTER XVI.

VALVES AND VALVE-GEAR.

Steam was admitted to and released from the cylinders of the early land engines by means of conical valves, operated by tappet gear in such a way that the steam valve was suddenly opened and as suddenly closed at the proper times, and the valve which allowed the steam to escape to the condensers and closed before steam was admitted worked with the same precision and by the same methods. Such an arrangement permits of a high state of efficiency for the steam, but is open to the objection that motion so sudden is liable to cause shock with much wear and tear of the working parts. In those early days the pressure of steam was only a little above that of the atmosphere, and the number of strokes per minute comparatively few, so that leakage past the valves was important, and the wear of the tappet gear not so great as might be expected; on the whole, the early engineers had every reason to be satisfied with their valve-gear.

That such gear can be made to work well and give general satisfaction is evident from the fact that it is still employed in large pumping engines, which work at 60 lbs. pressure and move at much higher speeds than formerly obtained, and in modern times in the marine type large electric generating engines at central stations, as well as in all oil engines.

Modifications of this form of valve and valve-gearing have been adopted for marine purposes, but not with that degree of success, at least in this country, as to commend themselves to engineers for extended adoption. In America, however, they are still largely employed for marine engines, and most successfully so in the various engines of river and lake steamers moving at comparatively low rates of revolution.

Such gearing was impossible in the locomotive engine on many grounds, so that an entirely different valve was adopted to suit it. This valve differed from previous valves in many ways, but chiefly in respect to its motion, which was a sliding one. From it this form of valve is called a *slide*-valve, and frequently, when in its simple form, *the locomotive slide-valve*, to distinguish it from the long and short **D**-valves and piston-valves so generally used in the engines on land and sea in the early part of the nineteenth century, which also had a sliding motion.

The locomotive slide-valve (fig. 135), in its simple and extended forms, is the one most generally used in the marine engine. It consists essentially of a rectangular block having a central cavity, the flat bars between the outer edge at each end and the central cavity being sufficiently broad to cover the cylinder ports, and when in its mean position both ports are covered. The amount by which the outer edges overlap the ports when in the mean position is called the *lap*, and the amount by which the valve is open when the piston is at the commencement of its stroke is called the *lead*; the space through which the valve is moved during half a revolution of the engine is called the *travel*. The amount by which the inside edges of the valve overlap the port is called the *inside lap*, and when, as often happens, instead of overlapping inside, the port is slightly open to the cavity of the valve, the valve is said to have *negative inside lap*.

The early locomotives made by Stephenson had, like modern steering engines, little or no lap, and little or no lead. Timothy Hackworth, by giving the valves of his engines lead and lap, effected an earlier cut-off, and consequently obtained some expansion of the steam, thereby saving the locomotive from threatened failure, and making it a commercial success. The effects of *lead* and *lap* will, however, be shown later on.

Seaward's Valves.—To get a more perfect arrangement of cut-off and release than is possible with one valve, Seaward fitted a separate valve and ports for steam and exhaust, and to avoid the large amount of *clearance* which such an arrangement would entail, he used four valves, a steam and an exhaust-valve for each end of the cylinder, and by placing the ports close to the ends of the cylinders, reduced the passages to a minimum. Each valve was simply a flat plate, and worked by cams on the shaft, and how-

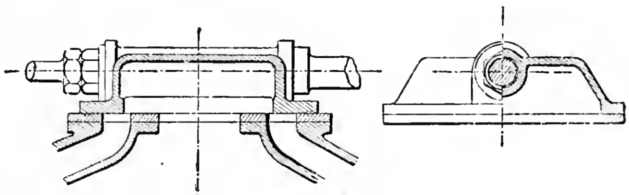


Fig. 135.—Common Locomotive Slide-valve.

ever early the steam-valves cut off, the exhaust-valves always opened just at the end of the stroke.

Common or Locomotive Slide-valve.—Fig. 135 shows the modern form of the valve, and it is so well known as to require no further description. So long as the cut-off is later than half the stroke of the piston, this valve answers very well for engines of moderate size; but when an earlier cut-off is required, sufficient opening to steam can only be obtained by excessive travel, early leads, or very broad ports, neither of which is desirable when it can be avoided. Large travel of valve means large power to drive it; broad ports produce the same result by increasing the area, and consequently the load on the valve; and early *leads* are apt to produce severe shocks on the rods, framing, etc., and to unduly check the piston.

Trick Valve.—Fig. 136 shows an ingenious plan for obtaining a double opening to steam by means of a passage around the valve, the entrance to which is at the end remote from that at which steam usually enters, and whose exit is through the lap or cover of the valve. It will be seen that the effective surface of the valve exposed to steam pressure is considerably reduced below that of the common slide-valve.

Double-ported Valve.—This valve (fig. 137) has a system of ports and passages which is added to the locomotive slide-valve, to allow of admission and emission of steam through a second port in the cylinder face, so that with the same travel as the common valve, there is double the area of opening for steam, and double the area for exhaust. This form of valve is very generally adopted for both the medium-pressure and low-pressure cylinders of compound engines of large size, and for the low-pressure cylinder of even quite small engines. Care must be taken in designing such a valve that

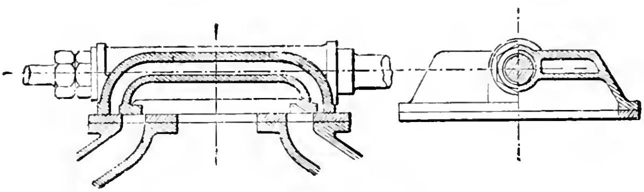


Fig. 136.—The Trick Valve.

there is the requisite area for steam to enter into the cavity leading to the steam port, and also that there is ample room at its back for the exhaust steam to pass from the outer port of the cylinder.

Treble-ported Valves (*v.* fig. 75).—To obtain still larger opening for steam, the lap of the double-ported valve is extended so as to cover a third port at each end; a portway is made through the cover lap, which admits steam

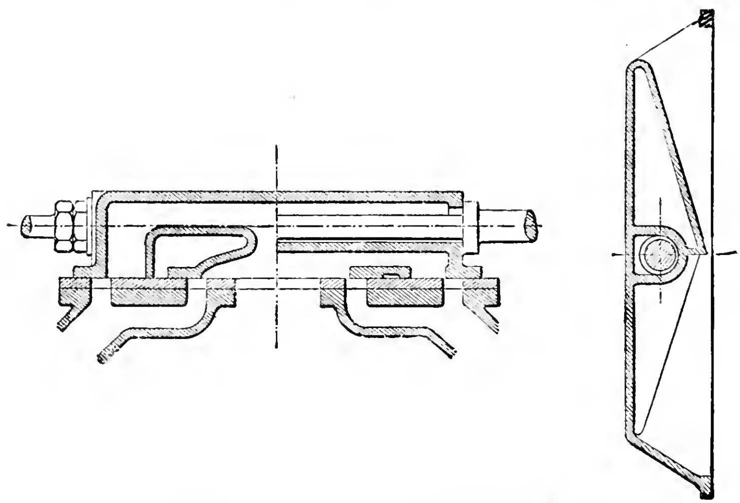


Fig. 137.—Common Double-ported Valve.

to the *second* port of the cylinder, while steam enters the third port of the cylinder through the opening beyond the valve in the usual way. But since the length of face and valve required for this form of valve is very nearly as much as if it had a port and passage to admit of exhaust, it has been discarded in favour of a treble-ported valve, similar in all respects to the double-ported. Some very large engines have four-ported valves, similar in design to the double-ported. Such valves are very large and heavy, and require a large amount of power to move them; but they work very satisfactorily, and so far have not any successful rivals for steam of low-pressure. When used for the medium-pressure cylinder of compound engines, both treble- and double-ported valves require relief plates or frames to diminish the large area they expose to steam pressure.

The Travel of Flat Valves should be such that their mean speed is not more than 200 feet per minute for the H.P. cylinder, nor 250 for L.P. cylinders.

Piston Valves.—No system of relief frames or plates has yet been tried which has given universal and entire satisfaction; some indeed produce more resistance than they have been designed to reduce, and the best cannot be depended on for any very long period when exposed to the temperature of high-pressure steam. The area of opening of port is restricted when only a common locomotive slide-valve is used, and its extensions magnify the evil which relief frames are supposed to cure. It has been stated that circular valves of the *mushroom* type do not work well in fast-running engines, although they give a good opening to steam. To combine the advantages of the two systems the piston valve is designed (fig. 138). The port area is nearly three times that of a flat valve of the same dimension transversely,

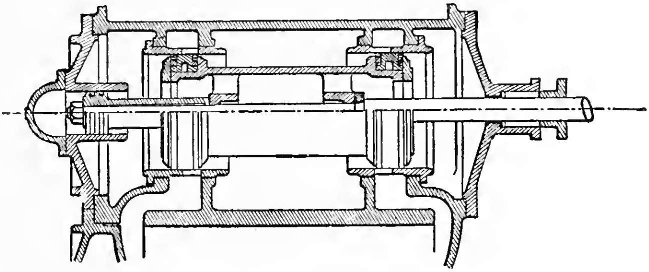


Fig. 138.—The Piston Slide-valve.

and the pressure on the sides due to the steam is nil. Essentially, the piston valve consists of two pistons, the face of each being equal in length to the bars of a locomotive slide-valve, and connected by a rod. These pistons are fitted into a cylindrical chamber having ports corresponding to those in the cylinder face; the faces of the piston cover these ports, and have the same amount of lap, etc., as a common valve. Steam is sometimes admitted outside the pistons, and it exhausts from the cylinder into the space between them, and from there into the exhaust passage in the usual way. But the piston valve permits of the steam being between the pistons and the exhaust steam passing their outside. In this case the valves are said to be with "inside cut-off." Piston valves were used very early in the nineteenth century, especially in marine engines, and found to be superior to the "Long **D**" and "Short **D**" valves; they were always of the "inside cut-off" type, and then, as now, had the advantage of confining the steam of high pressure to the circular casing surrounding the middle of the valve, so that no joint or stuffing-box was exposed to its action. This quality is to-day, with steam of really high pressure and high temperature, of great value, especially when the steam is rendered very dry by superheating, etc.

When the pistons are sufficiently large they are connected by a pipe or hollow casting (as shown in fig. 138), through which steam can pass from one end to the other; if this cannot be accomplished, the two ends of the valve-case are connected by a pipe cast with or connected to it.

Small engines, when fitted with such valves, have them in their simple form, the pistons being plain bronze or cast-iron discs of the required thickness, generally cast in one piece. Such a form would suit all sizes of engines, if always working at high speed; but when standing or running slow, the leakage past the valve, especially when it was worn, would soon be so considerable as to cause serious loss, besides making the engine very unhandy. To avoid this it is usual to pack the pistons much in the way that ordinary pistons are packed, except that the junk-rings and flanges are chamfered away, and the packing rings are made to project from them so as to allow free passage to the steam. The packing rings are also made sometimes with an extension so as to entirely mask the junk-ring and flange, so that the acting face is at least as long as the body of the piston portion of the valve. The spring-rings are made of strong cast iron, and since, owing to the very slight velocity at which the valve moves, the wear is small, the rings should have very little *set* or spring. When steam of high pressure is in contact with the valves of large size the rings should be restrained from pressing on the walls, as in the case of pistons (fig. 88). The liners in the valve-box are usually made of cast iron, fitted tightly in and secured by flanges. With superheated steam, and also to prevent rusting fast when not at work the packing rings may be of a very hard bronze like bell-metal.

There are *diagonal* bars across the ports to act as retaining guides to the packing rings; these bars are usually from $\frac{3}{4}$ to $1\frac{1}{4}$ inches broad, and take away about a third of the gross portway. The passage way around the liner must be so designed as to allow due area of section for the passage of steam; and to economise space, and reduce the clearance volume, they are eccentric to the liner and valve. To avoid the chief defect in these valves—viz., large clearance space—the valve should be long, so that its ports are nearly in line with those at the cylinder bore.

Piston valves are now very general, and prolonged experience of them has given the necessary confidence, even for their more extended use. For steam of a pressure over 100 lbs. they are a necessity. Some manufacturers use them for the low-pressure cylinder, but few engineers will care to incur the expense of them for a purpose where they are generally quite unnecessary. They are, however, necessary for the fast-running engines, such as fitted in torpedo boats, destroyers, and very high-revolution express steamers, etc., where safe continuous running is of more importance than extreme economy. They also have some advantages for very large engines, when the port area is so great that two valves are required to avoid cutting away and disconnecting the cylinder ends so much from the body.

Relief Frames.—The resistance of a flat slide-valve is almost wholly due to the friction on its face, and the greater the pressure on the valve the greater will be the friction. This is due to the fact that the pressure on the back of the valve is only very partially balanced by that on the front. The common locomotive slide-valve may, during certain portions of its stroke, have no pressure at all tending to press it off its face. Just at cut-off it is relieved, however, to the extent of the area of one port, but this relief decreases as soon as the steam in the cylinder expands: and if it exhausts before opening at the outer end for steam, its whole area is exposed again to the full amount pressing it to its face. It will be seen from this that the resistance is ever varying, and this condition is true also of double- and treble-ported valves, so that if a *definite and fixed* area of the back of the valve is so covered as not to be exposed to steam pressure, it does not quite meet the case; this, however, is the usual way in which the valve is relieved, the area being such that by the balance of pressure the valve is always pressed to its face. An exception to the rule occurs in the relief arrangement of the valves designed by Messrs. J. I. Thornycroft & Co. These valves have a face on their back similar to their front, with recesses corresponding to the ports; on the back of the valve there is a plate, which has ports like those

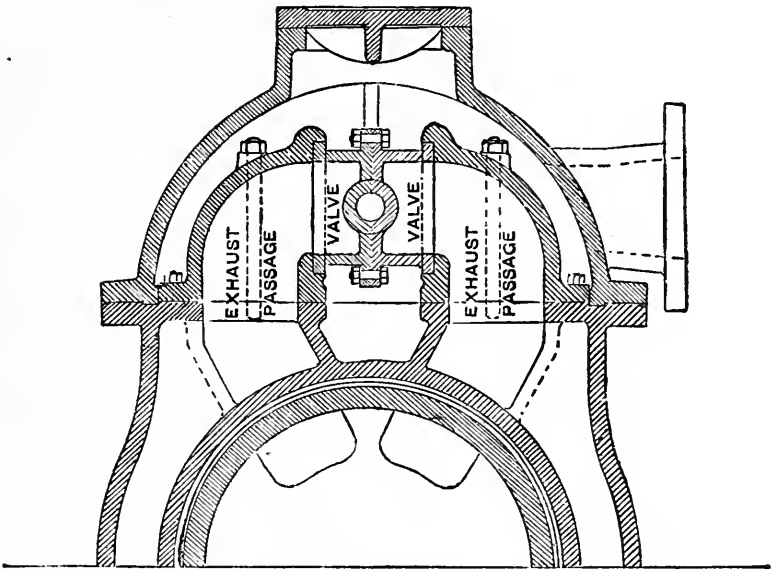


Fig. 139.—Self-balanced Valve.

on the cylinder face, bearing gently, in one respect, therefore, it is as evenly balanced as a piston valve.

Double Valves.—Fig. 139 shows an ingenious arrangement by which one valve is made to balance another by attaching one to the other, and as the two valves operate for the same purpose, each may be half the size of an ordinary valve.

The two valves are faced up, and so set that they are a slack fit between the two faces when cold. The port-ways are, as shown on the diagram, bolted to the main casting on each side, and from their form the pressure of the steam in the valve-box causes them to spring and so approach one another, making the combined valve tight on each face. At the same time, the pressure is not sufficient to cause very great resistance to the motion of the valve. This form of valve has been found in practice to work very well indeed, and many months of successful use have produced little or no wear on the valve faces.

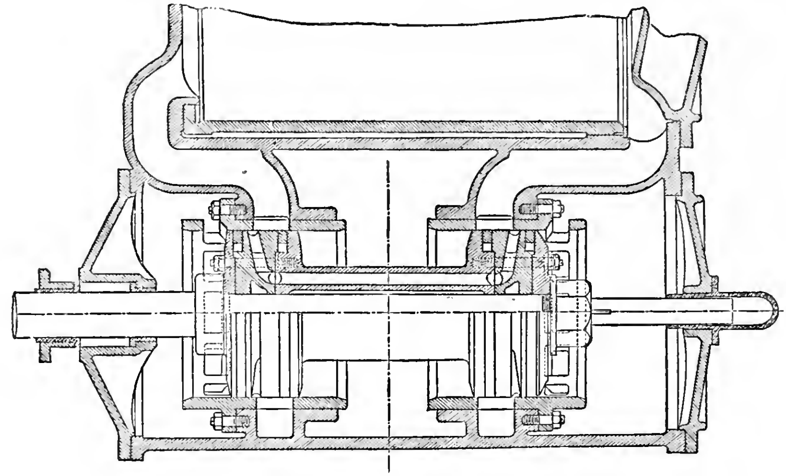


Fig. 140.—Thom's Patent Piston Valve.

Fig. 140 shows a piston valve adapted by Mr. Thom with the same principle as the Trick valve.

This arrangement rather complicates the piston valve and renders it far less simple than in the ordinary form, but there are no practical

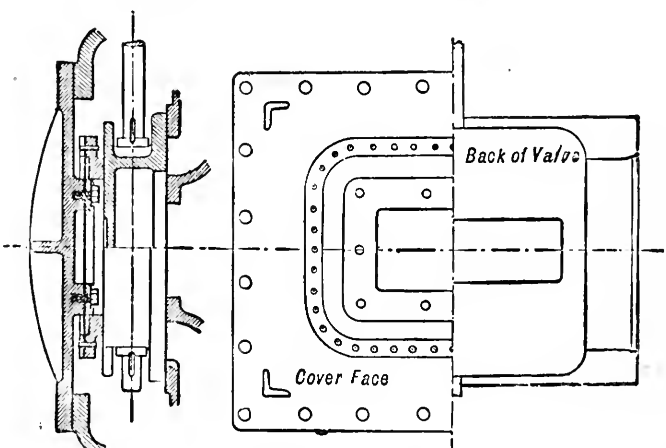


Fig. 141.—Dawe & Holt's Patent Relief Frame.

difficulties in the way of accomplishing what Mr. Thom aims at—viz., to admit some steam for the purpose of cushioning at one end of the cylinder from the exhaust just commencing at the other end. This, when quiet

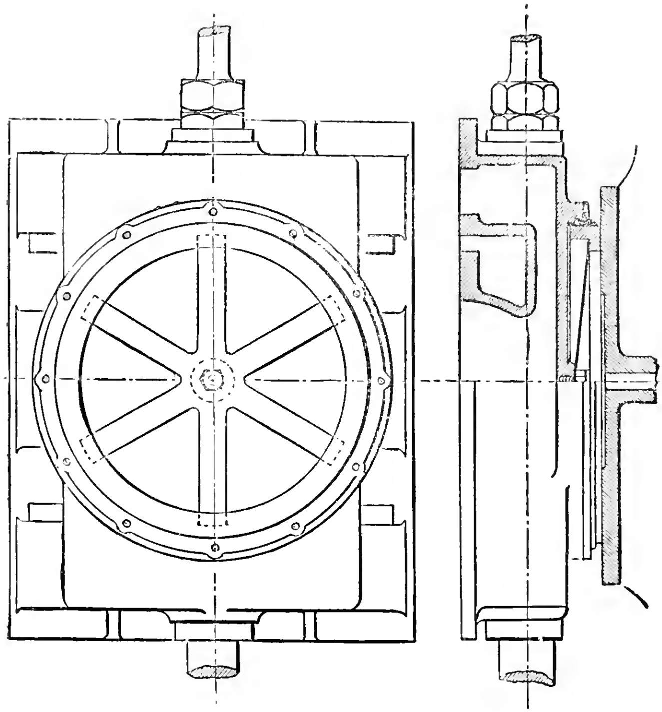


Fig. 142.—Common Relief Valve.

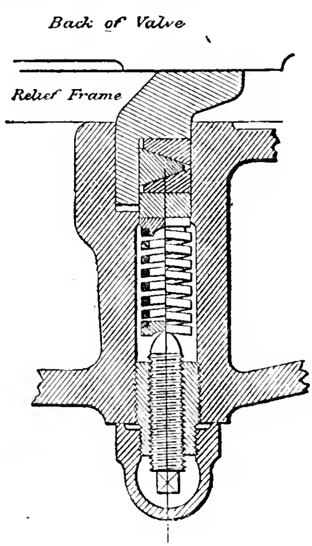


Fig. 143.—Spiral-spring Packing to Relief Frame

running is necessary, it is hardly needful to say, is of considerable advantage with the low-pressure cylinder, but it is not so necessary to the medium-, and not at all necessary to the high-pressure cylinder of a triple-expansion engine.

Dawe & Holt's Patent.—This consists of a rectangular cast-iron frame, fitting steam-tight to the face on the back of the valve, and riveted to a steel or bronze sheet, which is itself secured steam-tight to the valve (fig. 141). The diaphragm is of steel hard copper or phosphor bronze, and only

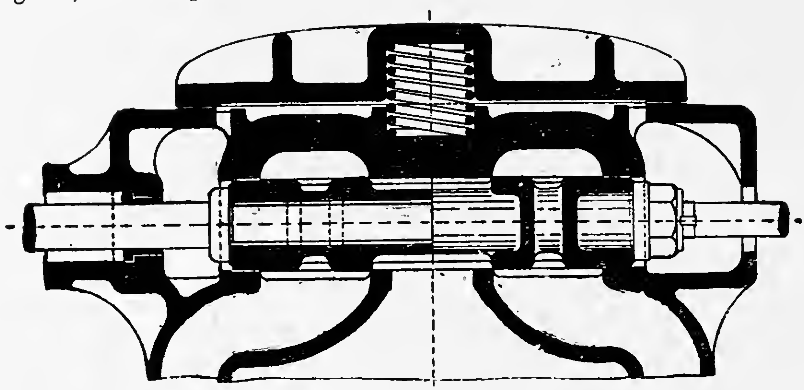


Fig. 144.—Martin and Andrews Valve Relief.

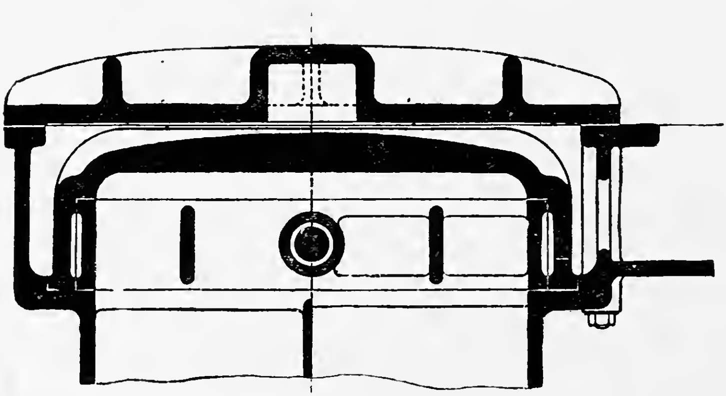
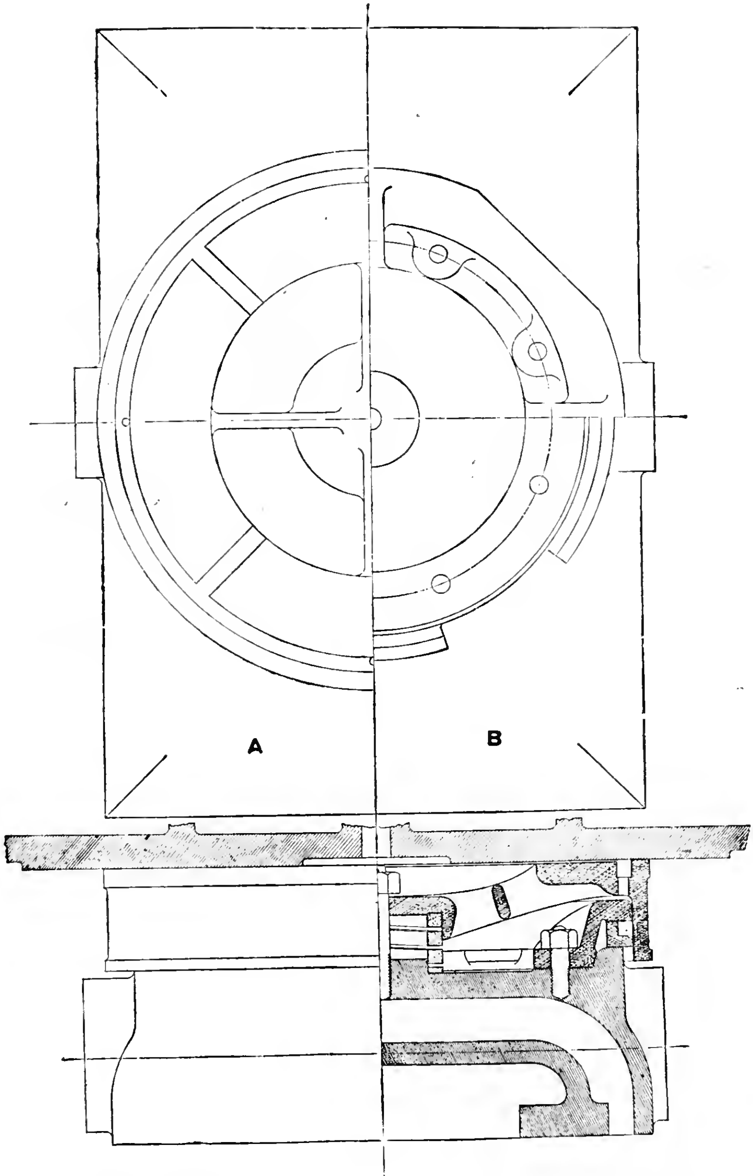


Fig. 144*a*.—Transverse Section of Martin and Andrews Valve.

18 B.W.G. thick, and so allows the frame to move slightly to suit the valve. This is so made that the frame is pressed back about $\frac{1}{16}$ inch when the valve is in place, and the frame itself, being exposed to steam pressure, is always pressed against the valve, the area enclosed by the frame being the amount of relief given to the valve.

Common Relief Frame.—The ordinary method of relieving the back of the slide-valve from steam pressure is by means of a frame of circular or rectangular form, fitting steam-tight, but freely, into a recess in the valve-



SIDE ELEVATION. | SECTION THROUGH MIDDLE OF VALVE.

Fig. 145.—Church's Patent Relief Frame.

A. Plan of Relief Ring. | B. Plan with Ring Removed.

box cover, and pressed against the smooth face at the back of the valve by set screws, an elastic material, or spiral metallic springs, as shown in fig. 143, being interposed between the screws and the frame, in order to give greater freedom to it.

Fig. 142 is a modification of the common plan, and is especially applicable to valves of nearly square form. In this case the relief frame is circular, fitted into a recess in the back of the valve, and pressed out by a star-shaped steel spring.

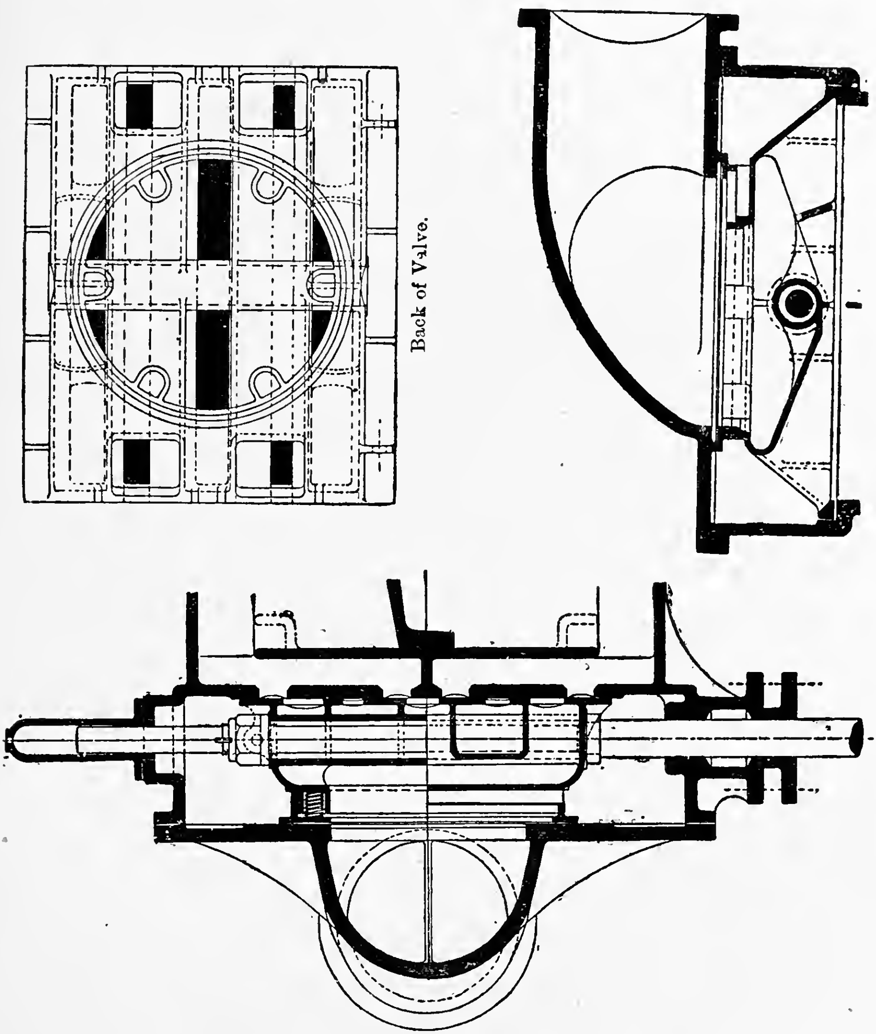
Martin and Andrews Valve.—Thornycroft's frame consisted of a casting with recesses and faces corresponding to the ports and bars of the cylinder face; the valve was with flats and recesses on its back corresponding to its face. The frame rested on legs touching the cylinder face, and just touched the valve back so as to keep the steam off the surfaces of contact. Figs. 144 and 144*a* represent a modification of this plan so improved that what appears as an ordinary single-ported loco valve really acts as a double-ported one by allowing steam to enter at both front and back edge, and to pass to exhaust at two places in the same fashion—viz., by way of the little port crossing from front to back of the valve at each end. This invention is patented by Messrs. Martin and Andrews, and much used by the former engineer for the cylinders of the large and successful quadruple engines turned out by him from the Royal Schelde Dockyard at Flushing.

There are many other plans for preventing slide-valves from pressing unduly on the cylinder face, some by means of springs, and some by pistons acting perpendicularly to the direction of motion of the valves, but none of them are free from objection, nor have any given greater satisfaction than Church's (fig. 145) very ingenious arrangement.

Through Exhaust Valve.—Fig. 146 shows a low-pressure double-ported valve arranged to exhaust through its back direct; provided the packing-ring is well designed, well made, and kept in good order, this is a safe and useful one, as it permits of the face being much closer to the cylinder than ordinarily, and a perfectly unobstructed lead to the condenser.

Back Guides and Springs.—To prevent slide-valves from being forced from the cylinder face, it is necessary to have guides behind the valves, with suitable rubbing surface on the valve itself; but as it is not always easy to provide such guides, and since, when the valve and face have worn somewhat, they fail to keep the valve to its face, it is generally preferable to fit a pair of flat bar springs, whose backs press on rubbing strips on the valve, and the ends on a fixed part, one end of each being secured, and the other, free to slide on the flattening of the arc.

Balance Pistons.—Since the weight of the valves, their rods, and gearing, in the case of vertical engines, is taken by the eccentric straps, and if it were unbalanced the top half of the straps would be subject to more wear than the lower, it is advisable to provide some means of avoiding this. The readiest is by fitting to the top end of the valve-spindle a piston (figs. 78 and 138), which works in a cylinder provided in the valve-box cover or top; this piston is of sufficient area that the steam-pressure on it affords the balance required, and it also acts the part of a guide for the spindle. When the eccentric-rods are very long, it is better to give an excess of area to the balance piston, so that the rods may be *always in tension*, instead of alternately in tension and compression. Since the pressure in the valve-box of the low-and medium-pressure cylinder of a compound engine is constantly varying it is better, especially when the above object is aimed at, to place the balancing cylinder outside the box, and supply steam from the boiler to the under side of the piston, so that the balancing force may be constant; but, on the



Horizontal Section.

Vertical Section.

Fig. 146.—Valve with Exhaust through the Back.

other hand, since the resistance of the valve varies with the pressure in the valve-box, there is not the same necessity for so much balancing force as would be the case were it otherwise; so that, on the whole, the piston exposed to the pressure of the valve-box is sufficient for the needs of most engines.

The top of the balance cylinder of the high- and medium-pressure valves should be connected to the next receiver by a small pipe, and that of the low-pressure valve to the condenser.

Joy's Assistant Cylinder.—In the hands of the late David Joy the balance piston with its cylinder has been transformed by a very ingenious and simple device, from merely passively supporting the weight of the valve and its gear, to actively aid the link motion in moving the valve up and down. Fig. 147 shows the method by which this was accomplished at the outset; here the steam is admitted to the top and bottom of the piston through

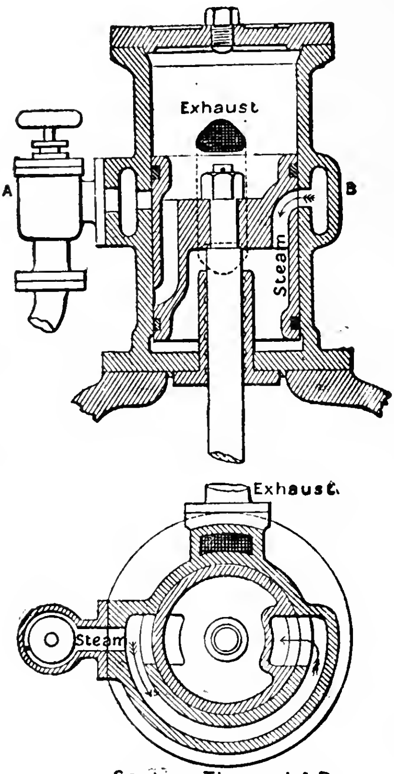


Fig. 147.—Joy's Assistant Cylinder.

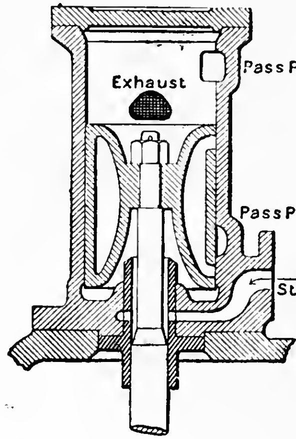


Fig. 147*a*.—Joy's Improved Assistant Cylinder.

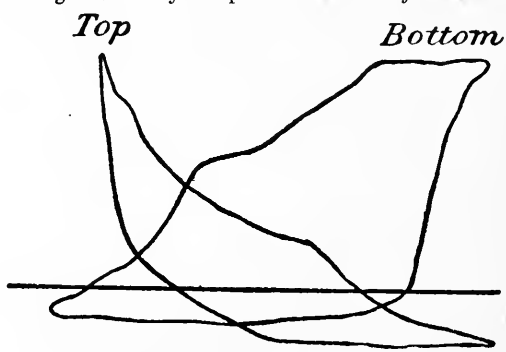


Fig. 147*b*.—Indicator Diagram from Joy's Assistant Cylinder (fig. 147*a*).

passages in the piston itself which open to steam ports in the sides of the cylinder at or near the end of the stroke. The exhaust is also through ports in the side of the cylinder which are opened and closed by the piston itself towards the end of the stroke. Fig. 147*a* shows another method, which Mr. Joy called the compound system, inasmuch as the steam contained in the body of the piston, after partially expanding on the upstroke, is passed by means of a recess near the top of the cylinder into the space above the piston so as to expand there and so impel the piston downward. The steam obtained access from the underside to the interior of the piston by means of another recess near the bottom. The valve spindle rod is reduced in diameter for some little distance below the piston, and, passing through a sleeve with port holes in it, does duty as a valve for the admission and cut-off of steam. Just before cut-off the piston passes the lower recess so as to allow its interior to fill. The piston face in way of the exhaust ports is, of course, plain, so as to completely cover them and prevent the steam inside the piston from passing to exhaust.

Fig. 147*b* shows a set of indicator diagrams taken from an assistant cylinder working under these conditions, and they certainly are remarkable, considering the simple method by which the steam is distributed and released. These assistant cylinders have been fitted by several firms to fast-running engines which have of necessity heavy slide-valves, especially those of the low-pressure cylinder. The results have been very satisfactory, and the headgoing, eccentrics, etc., relieved to a very considerable extent from the heavy work entailed in operating such valves at a high speed.

* **Valve-rods or Spindles.**—Since it is possible for a slide-valve to be exposed to the pressure of steam on its whole area, without any relief due to the ports, etc., and this may occur even when the valve is fitted with relief frames, it is better to assume this in making all calculations for determining the sizes of the parts to move it.

If L be the length of a valve, and B the breadth in inches, p the maximum absolute pressure to which it is exposed in pounds per square inch, then,

$$\text{Maximum pressure on the valve} = L \times B \times p \text{ lbs.}$$

The coefficient of friction should be taken at 0·2, or that of metallic surfaces rubbing together dry, as this is the worst condition likely to occur, then,

$$\text{Load on valve-rod} = 0{\cdot}2\,(L \times B \times p) \text{ lbs.}$$

Since the spindle has to take stresses similar to those on a piston-rod, it may be dealt with in the same way.

A stress of 3,000 lbs. per square inch should be allowed in the case of comparatively long rods, and 3,600 when very short and well-guided. Hence

$$\text{Diameter of slide-valve rod} = \sqrt{\frac{L \times B \times p}{F}}.$$

When the rod is long F = 12,000.
,, ,, short F = 14,500.

In the case of the valve of the low-pressure cylinder, the pressure should be taken at 25 lbs. absolute, as the pressure in the receiver is seldom as high as 15 lbs. above that of the atmosphere. For convenience, the rods for both high-, medium-, and low-pressure cylinders should be alike, and consequently the size should be the largest given by calculation from the above rules.

When the guides of the valve-rod are above the rod-end (*v.* fig. 148), so that in reversing the link a severe bending strain would come on the rod, it should be somewhat larger in diameter in the guide. In cases of this kind, F should be taken at 20 per cent. less than given by the above rules for that part.

* N.E. Coast Inst. E. and S. rule for valve spindles is $\frac{\text{diameter piston-rod}}{2} + \frac{1}{4}$ inch.

Valve-rod Bolts.—When the joint at the valve-rod end is made capable of adjustment by means of two bolts (fig. 152); then,

$$\text{Diameter at bottom of thread} = \sqrt{\frac{L \times B \times p}{H}} + 0{\cdot}25 \text{ inch.}$$

When the bolts are of mild steel, H = 48,000. This same rule applies

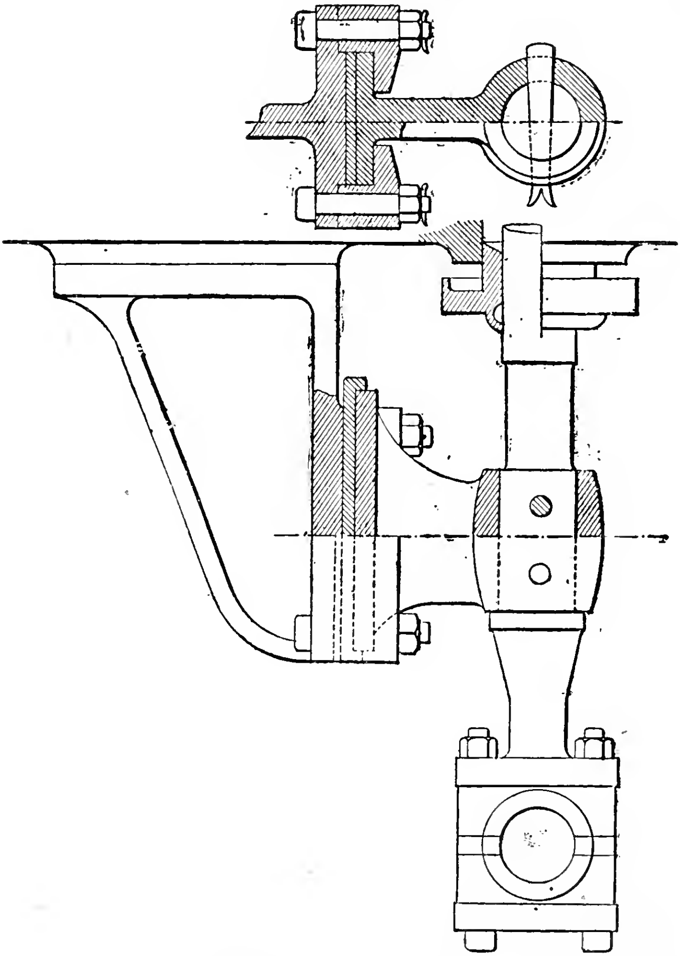


Fig. 148.—Valve-rod Guide.

to the bolts of eccentric-rod ends, and to the bolts at the butt-end of eccentric-rods.

For the bolts of eccentric straps, H = 40,000 for mild steel.

Valve-rod Guides.—The valve-rod end should always have a guide, so that the side strain due to the obliquity of the eccentric-rods, or to the action of the link motion, shall not only not bend the rod, but shall not come on the gland and so cause leakage, etc., through the stuffing-box. The kind of guide depends very much on the link-motion used, as what suits one form is not applicable when another kind is used. The simplest form consists of a cast-iron bracket with an eye bushed with bronze, through which the rod passes; but since the rod soon wears away the bush, if subject to much side pressure, this is not a satisfactory plan for large engines, although a

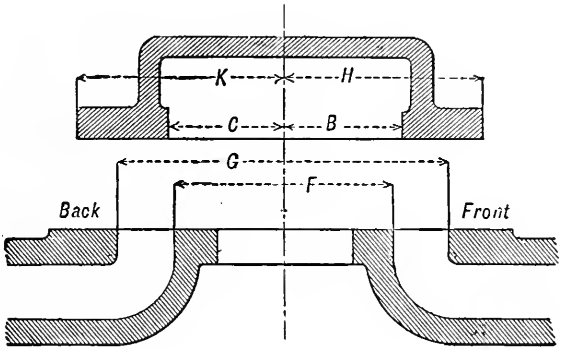


Fig. 149.—Proportions of a Common Valve.

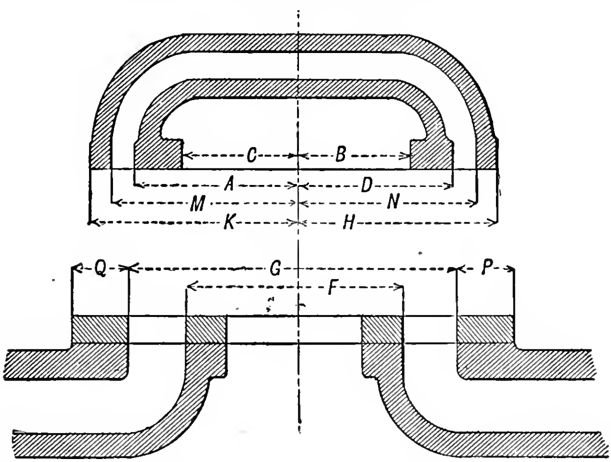


Fig. 149*a*.—Proportions of a Trick Valve.

convenient and fairly trustworthy one for small ones. If the rod is made of larger diameter in wake of the guide, and planed flat on the sides which are exposed to pressure, and the bracket fitted with brass liners, shaped like a common key, so as to be secured by screws through the gib ends, a simple and efficient means is provided for taking all side-pressure, but the drip from the gland is apt to wash out the oil. A better but more costly one is to fit a bracket to the valve-rod above the link, having a shoe like that of a piston-rod, and running on a guide like that of a piston-rod end. In large engines it is fitted with a slipper, etc., for adjustment (fig. 148).

Slot links require a different kind of guide when they work on overhung pins, which was often the case, of necessity, in horizontal engines, to obtain eccentric-rods of sufficient length. Then the valve-rod is fitted into the socket of a guide-piece, shaped somewhat like a bayonet, having a shank of square section and working in a guide bracket fitted to the framing.

Proportions of Slide-valves are found as follows:—

Let x be the *outside* lap of the valve at the front end, and y that at the back end. Then,

$$H = \frac{G}{2} + x; \qquad \text{and } K = \frac{G}{2} + y.$$

Let z be the *inside* lap at the front, and w that at the back. Then

$$B = \frac{F}{2} - z; \qquad \text{and } C = \frac{F}{2} - w.$$

To keep the valve as small as possible, E need be only a little wider than the steam ports, provided that the half travel does not materially exceed the width of bar.

The Double-ported Valve dimensions follow similar rules to the above.

Let x, y, z, and w be the laps as before. Then

$$H = \frac{G}{2} + x; \qquad \text{and } K = \frac{G}{2} + y.$$

Also,

$$B = \frac{F}{2} - z; \qquad \text{and } C = \frac{F}{2} - w.$$

$$A = \frac{G}{2} + \tfrac{1}{8} \text{ inch}; \qquad \text{and } D = \frac{G}{2} + \tfrac{1}{8} \text{ inch}.$$

The openings through the valve laps or covers must be as large as possible, but need not exceed the ordinary opening of the valve to steam at the outer edge; then

$$G + P = K + N;$$

and

$$G + Q = H + M.$$

Valve Gear: Link Motion.—The common form of motion employed to work the valves of a screw engine consists essentially of two eccentrics keyed on the crank-shaft in such a position relatively to the crank that when one is operating on the valve, the engine will propel the ship ahead, and is said to be in *head-gear*, and when the other, the engine will propel the ship stern first, and is said to be in *stern-gear*, their rods being connected by a bar or bars on which is a sliding block to which the valve spindle is attached. This bar connection is called the link, and is of such a form that by sliding it through the block, the *head* or *stern* eccentric may at pleasure be brought to operate on the valve.

In designing a link gear, the most important objects are to give the valve such motions as shall cause it to open to steam at or slightly before the piston is at the end of its stroke, the amount by which it is open at the end of the stroke, or commencement of the next stroke, being called the *lead* of the valve; then, to open fully, and close quickly at the required period of the stroke of the piston called *cut-off;* to confine the steam during he remaining portion of the stroke, that it may *expand* in the cylinder, and at or just before the end of the stroke to allow the steam to escape freely from the cylinder, called *exhaust;* to close the port again some time before the end of the stroke, so that the piston *compresses* the steam remaining in the cylinder and port, to form an elastic cushion for the piston, etc. These operations should be effected with the expenditure of as little power as possible, and with this end in view the motion of the link should be, as far as can be, limited to moving the valve only; consequently the link itself should have, when in full gear, no sliding motion longitudinally, called *slotting motion*, in the block, but only the angular motion due to the two eccentrics. A perfect valve motion is such that the valve opens to steam *wide* soon after the crank has passed the *dead centre*, and remains wide open during the admission of steam, so that there is no wire-drawing; the valve should close suddenly, and remain closed during expansion; at the end of the stroke it should open wide to exhaust, and remain in that state during the greater part of the period of exhaust, when it should close suddenly, and remain closed till opening again to steam. This is not obtainable with the ordinary link motion, nor to its full extent with any motion when one valve only is employed for both ends of the cylinder, because the period of cut-off at one end does not, as a rule, correspond to the period of compression at the other end; but there are valve gears which have two periods of very quick motion, and two of very slow in each cycle which very closely fulfils the above conditions, and which will be noted later on.

Slot Link.—This, which is one of the oldest forms of link, is still retained by many engineers, and is well adapted to the circumstances of several forms of engine, such as the oscillating paddle-wheel engine, and all engines in which there is not direct connection between the eccentric-rods and the valve-rod, as in some of the horizontal engines when it is either impossible or inconvenient to have direct connection. It is also much liked by those responsible for quick-revolution engines, as there are no nuts to slack back or bolts to break.

Fig. 150 is an illustration of the ordinary slot link, having means of adjustment for the sliding block and eccentric-pin brasses. Locomotive engineers prefer, as a rule, to have the pin-holes fitted with hard bushes rather than adjustable brasses; but this opinion is not shared by marine engineers, and chiefly on the ground that in a foreign port it is seldom possible and never convenient to engage the services of workmen and tools to renew these bushes when so badly worn as to require adjustment and therefore renewal.

This kind of link is generally suspended from the end next the head-going eccentric-rod, at a point in line with the arc through the block-pin; and if the pin in the lever, which operates on the link to reverse it, is placed in the proper position, there is very little slotting motion indeed when working in *head*-gear. The same remark applies to the position of the pin when in *stern*-gear, except that the amount of slotting motion is somewhat greater of necessity; but since a marine engine, as a rule, works very little in *stern*-

gear, and its efficiency there is of small consideration comparatively, this defect is of little moment. Locomotive engineers still keep to the slot link, as it has no small bolts and nuts to rattle loose and give trouble, as might be the case with other gears; but in the locomotive the eccentric-rod ends are often connected to the ends of link at the place where the drag-rod is in fig. 150.

Position of Suspension Pin.—To obtain the best position of reversing lever pin, it is necessary to delineate the path of the centre of pin in the link end through one revolution of the engine when the link has *no slotting motion*. The path so found is like an attenuated figure 8 in head-gear, and somewhat more pronounced in stern-gear. The arc of a circle of radius equal to the length of the suspension or bridle-rods, is then drawn through each of these figures in such a way that there is the least possible deviation

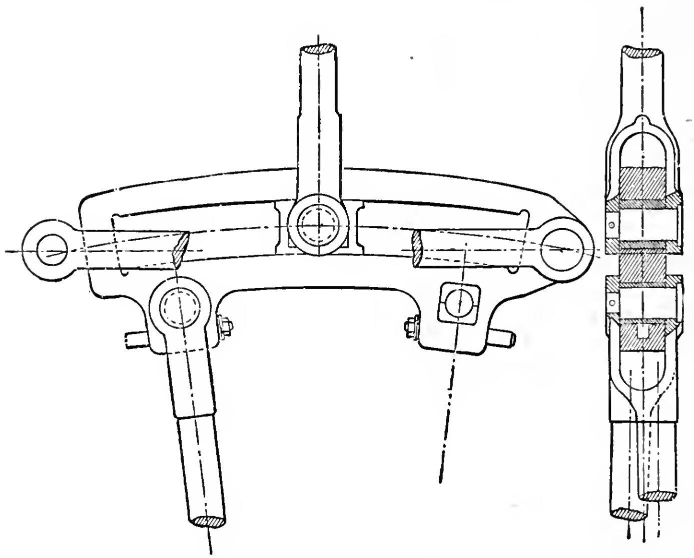


Fig. 150.—The Slot Link.

from the figure on either side—that is, the arc is the centre line of the figure, if the deviation on one side equals that on the other. The centres of the circles to which these arcs belong should be the centres of suspension of the bridle-rods, or position of pin in rever ing lever end. By drawing arcs of circles of radius equal to the length of the reversing lever from these two centres, the points of intersection are the two possible positions for the centre of weigh-shaft.

This same method of construction is suitable to all kinds of links, and for all positions of the point of suspension of the link.

When it is necessary that the valve motion shall be as satisfactory in *head*-gear as in *stern*-gear, the link should be suspended from a point in the line dividing it symmetrically, and by preference at the intersection of this line with the arc through the centre of block-pin, so that the centre of suspension is in line with the centre of block-pin when in *mid-gear.* When this is so, the pins for suspending the link are on side plates bolted to the sides of the link.

The distance from centre to centre of eccentric-rod pins should not be less than two and a half times the *throw* of the eccentrics, and is usually, when space permits, two and three-quarters to three times. The *throw* of the eccentrics in this case is, of course, equal to the travel of the valve when in *full* gear.

Size of Slot Link.—Let D be the diameter of the valve spindle, R being the revolutions per minute, and F = (13,500 — 10 R), from the following calculation

$$D = \sqrt{\frac{L \times B \times p}{F}};$$

then

Diameter of block-pin when overhung	. . .	= D.
,, ,, ,, secured at both ends	.	= 0·75 × D.
,, eccentric-rod pins	. . . .	= 0·7 × D.
,, suspension-rod pins	. . . .	= 0·55 × D.
,, suspension-rod pin when overhung	. .	= 0·75 × D.
Breadth of link	. . . .	= 0·8 to 0·9 × D.
Length of block	. . . .	= 1·8 to 1·6 × D.
Thickness of bars of link at middle	. . . .	= 0·7 × D.
If a single suspension-rod of round section, its diameter		= 0·7 × D.
If two suspension-rods of round section, their diameter		= 0·55 × D.

The objections to the slot link are, that it is an expensive one to make, and that, owing to the eccentric-pins and the block-pins being out of line, there is always an uneasy motion about the block-pin, and more slotting motion of the block. The former objection is valid, especially when the link is made of wrought material, but when the link is a steel or bronze casting it is not so expensive as some other forms. The uneasy motion is often due to bad design, for when well designed and carefully hung, it will work quite satisfactorily.

Single-bar Link.—This kind of link consists of a single solid bar, of rectangular section generally, and having the eccentric-rods connected to each end, and a sliding block between, to which the valve spindle is connected. The form of link, although tried by more than one eminent firm of engineers, was gradually dropped, except by the firm that introduced it.

Double-bar Links.—There are two kinds of double-bar links; one (fig. 151) having the eccentric-rod ends, as well as the valve-spindle end between the bars, so that the travel of the valve is less than the throw of the eccentrics; the other (fig. 152) has the eccentric-rods formed with fork ends, so as to connect to studs on the *outside* of the bars, and thus admits of the block sliding to the end of the link, so that the centres of the eccentric-rod ends and the block-pin are in line when in full gear.

The former plan is cheaper to make, is simpler in construction, and has fewer parts to get out of order and adjust; and when adjustment is required, it is easier to make, and there is less chance of its being done improperly. When in *head*-gear, part of the work of moving the valve is done by the *stern*-going eccentric, so that the wear is not limited to the one eccentric strap. When properly hung, the slotting motion is exceedingly small, and the valve motion is as perfect as with the other form. The objection to it is that the eccentrics are larger in diameter than those with the other links, and the links themselves are longer, and more space is required for the eccentric-rods to move in.

Size of Bar Links.—Let D be the diameter of valve spindle, found as before.

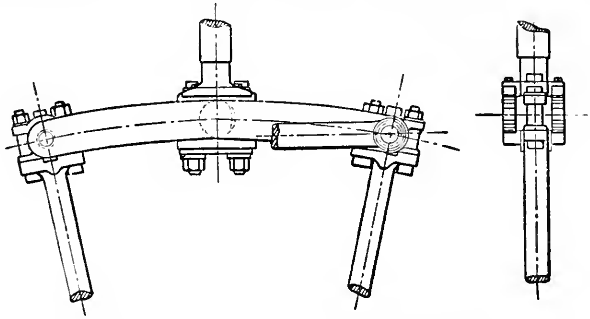


Fig. 151.—Double-bar Link with Rods inside.

Fig. 151, distance between centres of eccentric-pins, 3 to 4 times throw of eccentrics.

Depth of bars, ...	$= 1{\cdot}25 \times D + \frac{3}{4}$ inch.
Thickness of bars . . .	$= 0{\cdot}65 \times D + \frac{1}{4}$ inch.
Length of sliding block .. .	$= 2{\cdot}5$ to $3 \times D$.
Diameter of eccentric-rod pins .	$= 0{\cdot}8 \times D + \frac{1}{4}$ inch.
.. centre of sliding block .	$= 1{\cdot}3 \times D$.

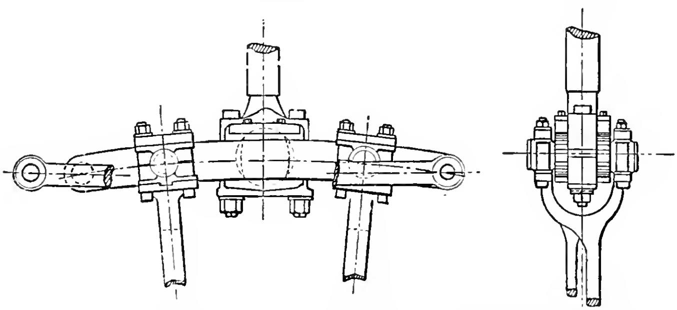


Fig. 152.—Double-bar Link with Rods outside.

Fig. 152, distance between eccentric-rod pins $2\frac{1}{2}$ to $2\frac{3}{4}$ times throw of eccentrics.

Depth of bars	$= 1{\cdot}25 \times D + \frac{1}{2}$ inch.
Thickness of bars . . .	$= 0{\cdot}65 \times D + \frac{1}{4}$ inch.
Length of sliding block . .	$= 2{\cdot}5$ to $3 \times D$.
Diameter of eccentric-rod pins .	$= 0{\cdot}75 \times D$.
Length ,, ,, .	$= 0{\cdot}80 \times D + \frac{1}{4}$ inch.

Diameter of eccentric bolts (top end) at bottom of thread $= 0{\cdot}4 \times D$ when of mild steel.

These bars should be of a very good description of iron and case-hardened, or of hard steel: the latter is, of course, free from seaminess and stronger. The eccentric-rod pins of the kind of link (fig. 152) are usually forged solid with the bars, but there is no absolute need of this, and it adds very much to the cost, both of manufacture and renewal when worn. Since the wear on these pins is limited to a very small portion of their circumference, it is not unusual to file away the parts which are not subject to wear, so as to admit of the brasses being closed when worn. When loose pins are fitted, they should be steel, and hardened, so that all wear may come on the brasses which are capable of adjustment.

In another arrangement of bar-link motion, the sliding block is divided, and on the outside, while the eccentric-rod ends are between the bars. This, while having some slight advantages, is on the whole very clumsy, and the block-pins wear badly; besides which, the link can be only suspended from the extreme end. Mr. Martin, of Flushing, adopts an ingenious combination of these bar links for the large engines he turns out. He fits three eccentrics to each valve; the rods of the two outer ones are connected each to a pin on the outside of the link, as in fig. 152, and act in "ahead" gear. The rod of the centre eccentric is connected to a pin between the bars, as in fig. 151, and acts in "astern" gear.

Single Eccentric Gear.—The valves of slow-working engines can be worked by a single eccentric, which is free to move round on the shaft from the position for *head*-gear to that for *stern*-gear; it is driven by a key or *stop* fixed to the shaft, pressing against a shoulder on the side of the eccentric. The eccentric sheave is balanced so that it will keep in any position, and only move when driven by the *stop* on the shaft. The eccentric-rod is so fitted that it may be disconnected from the valve-rod or its gear. This is generally effected by providing a *gab* or gap in the eccentric-rod end instead of a pin-hole, which allows the eccentric-rod to be lifted from its pin by suitable gearing. When the eccentric-rod is disconnected from the valve gear, the valve ceases to move, and the engine comes to rest. To restart the engine the valve must be worked by hand until the gab can be brought in line with the pin; if the motion of the engine is reversed the shaft will move around, the eccentric remaining motionless until the stop on the shaft comes in contact with the other side of the projection on the sheave; it is then in the right position for driving the valves, and the eccentric-rod may now be dropped into gear. This method admitted of the paddle engine being handled very dexterously when the valves were not so large and the steam pressure so high as to prevent their being moved by hand, and was the one generally adopted in engines of moderate power long after link motion was invented and in general use on locomotives.

There have been many other methods of moving a single eccentric from ahead to astern position; some of which by sliding wedges were at one time received favourably, but are now almost forgotten.

Another favourite method is that shown in fig. 69, where by means of a spiral slot in the eccentric sleeve and a straight one in the shaft a sliding bolt in the latter is made to twist the eccentric round.

Hackworth's Dynamic Valve Gear.—The locus of a point on a rod, one end of which moves in a circle, and the other on a straight line passing through the centre of that circle, is an ellipse whose major axis coincides with the straight line. If, however, the end of the rod slides on a line inclined to this centre line, the major axis of the ellipse will be inclined.

Hackworth's valve motion worked on this principle, for this gear had a single eccentric keyed to the shaft *exactly opposite to the crank*; the eccentric-rod had its end attached to a block, which slid on a guide-bar inclined to the line through the centre of shaft; and a rod from about the middle of the eccentric-rod was connected to the valve-rod, which works in a line at right angles to the line through the eccentric-rod when in mean position. If the guide-bar on which the eccentric-rod block slides be moved so as to reverse its inclination, the inclination of the axis of the ellipse was reversed, and the motion of the engine reversed. Here the *motion* of the valve is due to the eccentric, and the *lead* to the inclination of the slide-bar.

The great advantage of this gear, beyond the saving of an eccentric, is the better motion imparted to the valve, inasmuch as there are two quick and two slow motions in a revolution; the quick ones occurring at opening and cut-off, and the slow ones during entry and at exhaust previous to opening. A large variation in the amount of cut-off is possible with this arrangement, without wire-drawing from small opening and slow closing of the port, as is the case with the common link motion. The chief objections urged against this gear are the excessive friction, and consequently wear on the sliding blocks, and the liability of so many pins to derangement. The first of these is the most valid, and it has been overcome by fitting rollers instead of sliding blocks. In both ways, however, this gear has worked fairly well; and for engines of small power it is a very convenient arrangement, especially when much variation in cut-off is required.

Since the valve-rods with this gear do not come in line with the piston-rods, or immediately over the line of shafting, the valves and their chests are removed from the usual positions, so as to admit of the cylinders being closer together; the engine is consequently shorter, cheaper, and occupies less space. This suited the expansive screw engine and paddle engines of the diagonal type admirably; for the two valves of an expansive engine being in one common chest, could be examined through a door in front, of ample dimensions to admit a man; for the compound engine the valve-boxes are side by side with short-pipe connections (*vide* fig. 21); but generally it allows of little space for the receiver, and the exhaust from the low-pressure cylinder of screw engines must pass through a belt within the receiver, or else this belt has to traverse more than half the circumference of the cylinder.

Marshall's Valve Gear is a modification of Hackworth's, and differs from it in the method of getting the oblique motion of the rod end. Fig. 153 shows the plan adopted by the late F. C. Marshall, and also illustrates generally what has been said of Hackworth's. Here the eccentric-rod is hung, by means of a rod from the end of a lever on a reversing shaft, in such a way that it moves on the arc of a circle inclined to the centre line. The motion is not quite so perfect as with the inclined sliding bar, and necessitates double ports to the bottom end of the slide-valve, in order to get as much opening to steam there as at the top end; but there is less friction, and on the whole, it works most satisfactorily. The pins require to be of good size, and they should all have adjustable brasses to provide for the large amount of wear which of necessity comes on them.

Joy's Valve Gear.—Fig. 154 shows the ingenious arrangement whereby the late David Joy has avoided altogether the use of eccentrics. He obtained

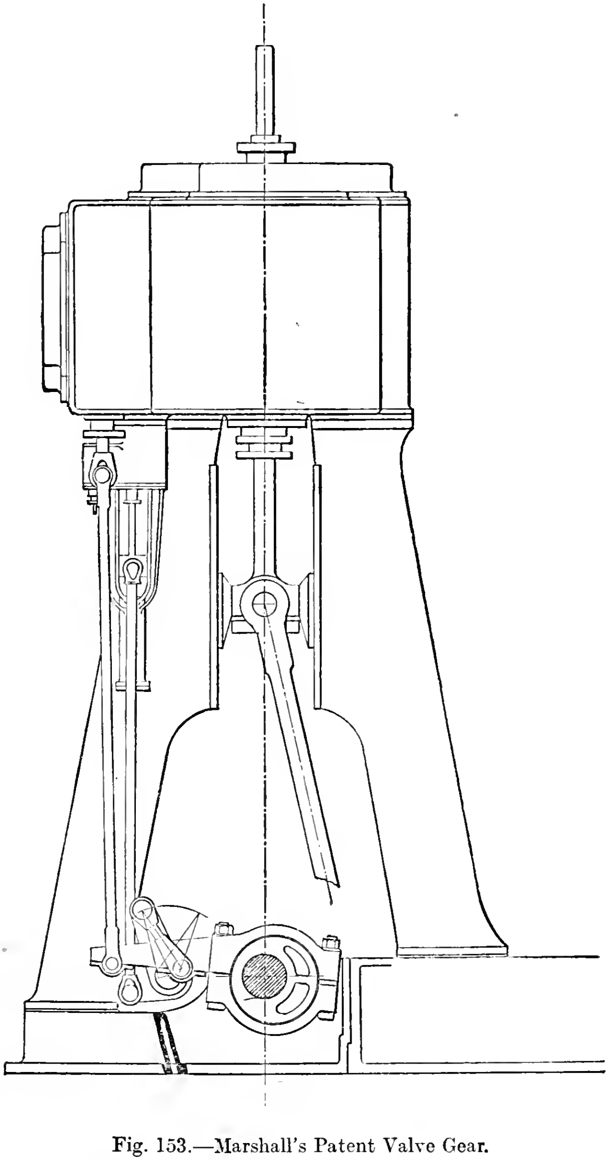


Fig. 153.—Marshall's Patent Valve Gear.

the motion from the connecting-rod and qualified it by a sliding block like Hackworth, or a suspension-rod like Marshall. The motion thus obtained is a very perfect one for a slide-valve, as the two quick and two slow periods are just when required, the amount of opening is equal at both ends of the valve, and early cut-offs can be effected without excessive leads and compressions or premature exhaustings. This gear has been applied with great success to locomotives, where the saving of space for the eccentrics admits of longer crank-pins and journals; it was also taken up by a few marine engineers. The chief objections to this gear are that it comes in the way of the principal working parts, and makes them a little difficult to get at when working, and also that a small amount of wear on the joints of the gear would cause a defect on the valve motion, and produce a serious amount of rattle of the gear. These, however, can be got over by making the pins substantial, and all the joints adjustable.

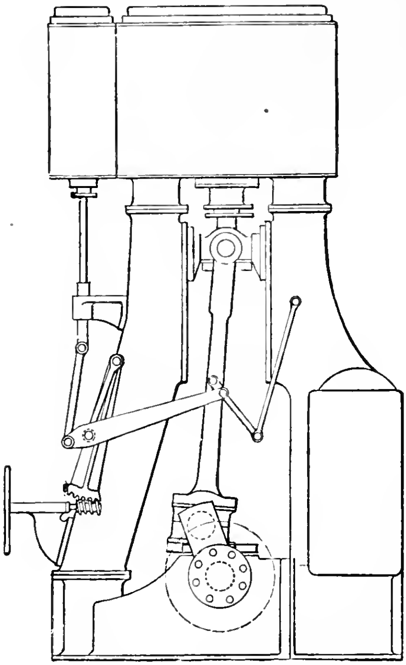


Fig. 154.—Joy's Valve Gear.

Sell's Valve Gear.—Here the valves are worked by an independent shaft, which derives its motion from the main shaft by means of wheel-gearing; between these two shafts are two intermediate wheels gearing into one another—one of them gears into the wheel on the main shaft, and the other into that on the valve shaft. These two intermediate wheels are carried on a frame which can be lifted up and down, and is guided in such a way as to keep the wheels always in proper gear with their corresponding wheels. If the gearing is set so that the valves are moved by their crank-shaft, and drive the engine *ahead* when the frame is at the bottom of its traverse, the moving of the frame to the top of its traverse so alters the relative angular position of the shafts, that the valves will be set to drive the engine astern. This is easily understood by supposing the engine at rest, and the frame raised; the wheel geared into the one on the crank-shaft, in moving round the shaft also turns on its own axis, and in so doing turns its companion-wheel, which turns the wheel on the valve shaft, and consequently the valve shaft itself.

This gear was fitted to many engines in the British and foreign navies, and was especially suitable for the three-cylinder horizontal engines for which it was originally designed. It had, however, the objectionable feature of wheel-gearing, which, when new, made a great noise, and was liable to accident from *racing* of the engines, but when worn, these are magnified very considerably by the *back lash* which results. Had the wheels been machine-cut accurately to shape, with helical teeth of sufficient size, this gear might have been in use to-day, as it was a very convenient one for horizontal engines, as also it is for large vertical ones. A similar arrangement has been used by certain engineers for vertical triple engines, but the auxiliary or valve shaft was in this case driven by an auxiliary engine, which had a sensitive-ball governor to regulate it. In rough weather the wheels connecting the two shafts were thrown out of gear, and the main engine was said to be satisfactorily governed by the small one controlled by its own ball governor.

Expansion Valves.—When ordinary link motion is employed, and a much earlier cut-off than half-stroke is required as a normal condition, it is necessary to effect it by means of an independent valve, usually called the expantion valve. These valves are generally common plates, sliding either on a face with ports on the side of the valve-box, or on a face on the back of the ordinary slide-valve. In the latter case there are two methods, commonly known as *the inside cut-off* and *the outside cut-off*.

Gridiron Expansion Valves.—When the valve worked on an independent face on the side of the valve-box, it consisted of a plain plate with steam ports in it, corresponding to the steam ports on the face; there was sufficient lap, and the gear was so set that it remained closed after it had cut-off, until after the main valve was closed to the cylinder, when it might open again so as to fill the valve-box, and be ready to supply and cut off steam from the other end of the cylinder. The variation in cut-off is effected by varying the travel, and the equalisation of cut-off at each end is effected by varying the proportion of lap on each side of the ports. The variation in travel is obtained by means of a link on which the block to which the expansion eccentric-rod is attached slides, so as to vary the length of lever; the larger the travel of the expansion valve, the quicker is the steam cut off, and the smaller the travel, the later is the cut-off, until, finally, the travel is so small that the expansion valve does not close at all. The variation in the lap of the expansion valve is effected by means of the curvature of the link.

Some engineers preferred the mean position of the expansion valve to be with the ports closed; in this case the quickest cut-off is effected with the least travel, and consequently with least opening; but since the piston speed is comparatively slow when receiving steam with the early cut-off, a reduction of opening will not be felt; whereas the slow cutting-off when carrying steam to mid-stroke, effected by the valve, when the late cut-off is with least travel, produces considerable wire-drawing.

If the engine is generally to work with a cut-off at from $\frac{1}{4}$ stroke to $\frac{1}{2}$ stroke, then the former valve arrangement is the better; if the cut-off is to be, as a rule, from $\frac{5}{8}$ to $\frac{3}{8}$ the stroke, then the latter is better, except that it is somewhat inconvenient to have the ports closed when the valve has least travel, and also that when the expansion valve is not required it must be moving at its highest speed.

The chief objection to this kind of expansion valve is the large amount of clearance space between it and the piston, for the steam expands in the valve-box, as well as in the cylinder, until the main valve closes. It was, however, very generally used in the Navy for the cut-off for cruising speeds, and had the advantage of not interfering with or in any way affecting the main valves in case of anything happening to it.

Outside Cut-off Valves.—When the expansion valve works on a face on the back of the main valve, the latter is provided with steamways through it from the ports on its back to the ports on its face. If the main valve is single-ported, it is like a common locomotive slide-valve with enclosing ends, so that steam passes to the cylinder only through the passages thus formed. There are then two ports on the back of the valve, one of which is covered at cut-off by the expansion valve, and remains covered until the main valve cuts off. The expansion valve may consist of a single flat plate, which, when in mean position with respect to the main valve, is between the ports on the back of the main valve. If any variation of the cut-off is required in this case, it is effected by varying the travel of the expansion valve with respect to the main valve, or by altering the *lead* of the expansion eccentric. The latter plan would generally be very inconvenient, and, consequently, when such a valve is fitted, the variation is effected by varying the travel.

On account of the cut-off being effected by the outside edges of the expansion valve and the ports in the main valve, these are called outside cut-off valves. Since, however, the motion of the valve will be slow when cutting off with a small relative travel, it is unusual to fit a single plate, but to divide it and arrange the gear so that the two plates may be moved apart from one another so as to virtually increase their lap, and cause them to cut off earlier with the same travel. The result of this arrangement is that the expansion valve moves always at the same velocity and through the same space; the cut-off occurs when moving at or near the maximum velocity, whatever be the period of cut-off, and there is consequently little or no *wire-drawing* at the ports. The plates should be so designed that when cutting off at the earliest required period, they do not overrun the port so as to pass steam through at the *inner* edges, and that they may approach close enough together as to allow of a late cut-off if required.

The usual method of altering the position of the plates is by securing them by nuts to a common spindle, on which is cut a right-handed thread for the one plate, and a left-handed thread for the other, so that when the spindle is revolved the plates move in opposite directions. The nuts should, of course, be made of bronze, and provided with collars or other suitable device, so that when the spindle is turned round they do not revolve with it, but only move the plates. The valve spindle usually passes through both ends of the valve box, and is connected to the eccentric-rod by a swivel-joint, which permits it to be turned round; the other end has a featherway cut in it, and passes through a socket, which is held in place by a suitable bracket, provided with a fixed feather which fits into the groove in the spindle, and a wheel by which it can be turned; by these means the spindle can be turned while the engine is at work.

This plan suffices for small engines, but the power required to move the spindle rapidly increases with the size of the engine, and this not so much owing to the friction of the valves or external fittings, as to that of the screwed parts of the spindle in the nuts on the back of the valve from want of lubrication. This difficulty is so great that in engines of 50 N.H.P., only after a few days of work without altering the position of expansion valve, it has been found impossible to turn the spindle round with the means at command. To avoid this difficulty the gear for varying the cut-off should be outside the valve-box, and under the inspection of the engineer. A good plan for carrying out this is to provide each valve plate with a separate rod; each rod passes through a stuffing-box, and is connected to a cross-head by a nut in it, which is free to turn round so as to adjust the position of the spindle. These nuts may have wheels keyed to them gearing into one another, or they may have worm-wheels operated on by a worm between them; in either case the nuts will turn round in opposite directions, so that if the threads on both the spindles are right-handed, the turning of the nuts will cause the valves to move in opposite directions. The spindles are secured to the valve plates, so that they cannot turn round when the nuts are turned; they are also of necessity out of centre with the valves, which must consequently be guided sideways on the main valve. The only objectionable feature in this arrangement is, that the spindles are not secured to the middle of the valves; but practice has shown that this is no detriment, and the whole system works exceedingly well.

If the engine to which this kind of expansion valve is fitted, is required to work with the same efficiency in *stern-gear* as in *head-gear*, the eccentric should be in line with and on the side of the shaft opposite the crank, and its throw should not be less than that of the main valve eccentrics. If, however, the engine will seldom, and for only short periods, work in *stern-gear*, the eccentric should be on the same side as the crank, but nearer the *stern-going* eccentric than the *head-going* one; then the relative travel of the valves is greater in *head-* than *stern-gear*, and the cut-off is also earlier in head-gear, so that if a sudden order were given to *astern*, the engine is better prepared for carrying out that order.

The equalisation of cut-off at top and bottom of cylinder, for all positions of cut-off, is effected by causing one plate to move more per revolution of the spindle than the other; and in the case of the double spindle arrangement, the wheels on the nuts are so proportioned as to obtain the same result.

Inside Cut-off Valves.—So called, because steam is cut off at the inside edges of the expansion valve. The eccentric is in this case, also, nearly in line and on the same side as the crank, so that the relative travel with the same throw of the eccentrics is greater than in the case when the expansion eccentric is opposite the crank, and consequently on the same side as the eccentrics of the main valve.

The variation in cut-off can, in this case, be effected in a manner similar to the last; but since the relative travel is so much greater, and since it would be inconvenient to spread out the valve so much, it is usually effected by varying the travel by means of a link, as already described, or by a link having one end connected to an eccentric and the other to a rod jointed to a fixed point on the foundation, or more usually on the shaft, and called the *dumb* rod, because of its similarity in position to the eccentric-rod, without the longitudinal motion of the latter.

Piston Expansion Valves.—When the main valve is a piston valve, the expansion valve may be also a piston valve working within the main valve, and on the same principles as described for slide valves. This plan has been put into practice by several engineers, and although somewhat complex in structure, does, no doubt, work satisfactorily, and with little friction. The spindles are sometimes out of centre; but not always, as the expansion valve-rod passes through the top of the valve-box, while the main valve-rod is through the bottom as usual.

Expansion Valves for Compound Engines.—It has been stated that to effect a cut-off earlier than half-stroke *efficiently*, a separate cut-off valve is necessary, but this is only strictly true when the full speed of the engine is maintained with such a cut-off. If full speed is attained at a cut-off somewhat later than half-stroke, a *reduced* speed can be obtained *with efficiency* with an earlier cut-off by simply "linking up;" for the port area being suited to the higher speed, the reduction of opening on "linking up" is not materially felt at the reduced speed, the consequent compression also adds to the economic working of the engine at high grades of expansion, and if the valves have been so set that at full speed the work is evenly divided between the cylinders, so it will be, with but slight variation, on "notching up" the links of all the cylinders (*v.* fig. 155). Expansion valves are needless additions to the modern marine engine, and should be avoided, as all sources of possible breakdown or derangement ought. These valves have never been fitted to triple-expansion engines.

Eccentrics.—The sheaves, or, as they are sometimes called, pulleys, are made usually of cast iron, and, when possible, each is in one piece, bored out so as to fit the shaft on to which it is keyed. It is essential that it shall fit the shaft quite tightly, or otherwise it will soon become loose from the continual sudden and intermittent application of the load. When the couplings or flanges on the shaft do not admit of the sheaves being fitted in this way, it is usual to divide them on a line through the centre of the shaft at right angles to the line passing through the centres of shaft and eccentric. The two unequal parts are securely bolted together, and keyed to the shaft on the line through the centres, so that the whole of the strain comes on the larger and stronger half, the lesser half acting only the part of a clamp to hold it to the shaft. When it is desirable to keep the diameter of the sheaves as small as possible, this clamp piece should be of steel. Great care should be exercised in fitting the two parts of the sheave together, and also in "bedding" them on the shaft. Some engineers make all eccentric sheaves in parts, owing to the difficulty of fitting them tightly to the shaft when in one casting. When the division is made through the centres there should be a connecting bolt as close to the shaft as possible, as well as one at the extremity, and also there should be a key to each half of the sheave and the shaft. When the eccentrics are very large, and have to drive heavy valves, it is also advisable to make the small part, when divided, as above described, of wrought or cast steel.

The diameter of an eccentric sheave = 1·2 (throw of eccentric + diameter of shaft).

Breadth of the sheave at the shaft . .	$= 1{\cdot}25 \times D + 0{\cdot}65$ inch.
,, ,, ,, strap . .	$= 1{\cdot}2 \times D + 0{\cdot}63$,,
Thickness of metal around the shaft . .	$= 0{\cdot}7 \times D + 0{\cdot}5$,,
,, ,, at circumference . .	$= 0{\cdot}6 \times D + 0{\cdot}4$,,
Breadth of key	$= 0{\cdot}7 \times D + 0{\cdot}5$,,
Thickness of key	$= 0{\cdot}25 \times D + 0{\cdot}5$,,
Diameter of bolts connecting parts of strap	$= 0{\cdot}6 \times D + 0{\cdot}1$,,

The headgoing sheave should be 33 per cent. broader.

D is found as before, and is $= \sqrt{\dfrac{L \times B \times p}{F}}$ (*v.* p. 425).

	Steam.	Vacuum.	Revs.	Mean Pressure.		Indicated Horse Power.		
				H.P.	L.P.	H.P.	L.P.	Total.
1st Grade,	87	$25\frac{1}{2}$	73	59·92	11·62	430	379	809
2nd Grade,	90	$26\frac{1}{2}$	64	35·8	9·07	256	259	515
3rd Grade,	88	27	39	18·0	4·57	78·3	79·5	157·8

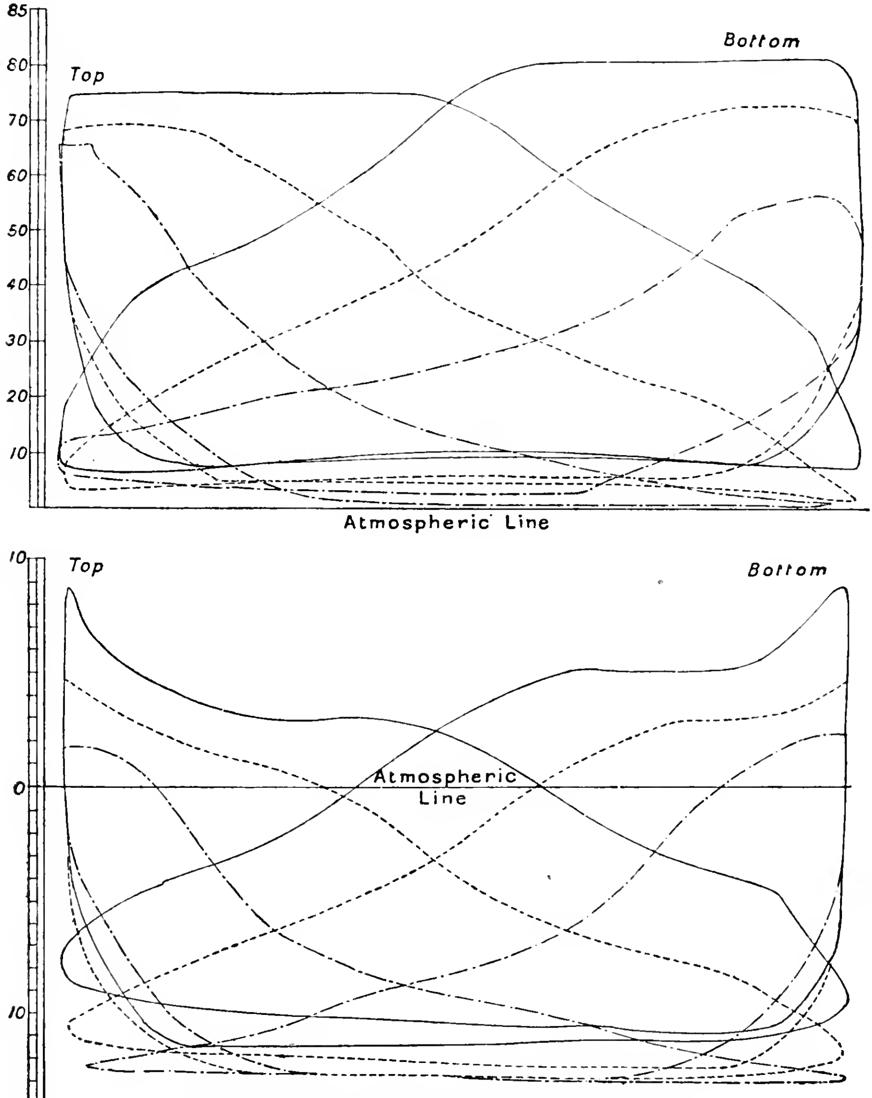


Fig. 155.—Effect of Notching up Links Simultaneously

Eccentric Straps are made generally in the form of a hoop, with lugs to connect the halves together, and with a base for the rod, if it is not formed with the strap. When the face of the sheave is formed with small flanges on each side, the section of the strap is rectangular; but the more common practice is to make the strap with small flanges, so as to overlap the face of the sheave, and bear on parts on each side. The advantage of the latter plan is that the sheave is not necessarily broader than the strap, and the oil does not run off so readily when the engine is at work.

Some engineers used to turn the face of the sheave with a **V**-shaped groove, and form the strap to fit into this; but it is not a good plan, owing to a small amount of wear permitting of very considerable side play (this again, though generally a fault, becomes a virtue with badly-fitted link-motions).

Eccentric sheaves have been made with a narrow projecting collar in the middle of the face, the straps having a groove turned to suit it. By this plan the straps are prevented from having side play, but the oil is not retained, as is the case with all the other methods.

The straps are usually made of bronze, or of wrought steel lined with bronze or white metal; but cast-iron straps give very great satisfaction, especially when of large size; cast-iron and steel castings lined with white metal have been adopted by some engineers with success; the latter, while having a strength beyond that of the cast iron, do not cost nearly so much as bronze or wrought steel, and are now being largely employed in the mercantile marine.

When of bronze or best cast iron :—

The thickness of eccentric strap at the middle $= 0{\cdot}4 \times D + 0{\cdot}6$ inch.
„ „ „ sides $= 0{\cdot}3 \times D + 0{\cdot}5$ inch.

When of wrought or cast steel :—

Thickness of eccentric strap at the middle $= 0{\cdot}4 \times D + 0{\cdot}5$ inch.
„ „ „ sides $= 0{\cdot}27 \times D + 0{\cdot}4$ inch.

Eccentric-rods.—Unless these are very short indeed, it is better to make them separate from the straps, as then the breakage of either does not condemn both. They are made of wrought iron or steel.

The diameter of the rod in the body and lower end may be calculated in the same way as that of a connecting-rod, the length being taken from centre of strap to centre of pin.

The diameter of eccentric-rod at the link end $= 0{\cdot}8\ D + 0{\cdot}2$ inch.

Eccentric-rods are, however, often made of rectangular section, which is the correct form for the stresses it has to withstand, but it is not so economic to manufacture.

Reversing Gear should be so designed as to have more than sufficient strength to withstand the resistance *both of the valves and their gear at the same time* under the most unfavourable circumstances; it will then have the *stiffness* requisite for good working.

Assuming the work done in reversing the link-motion, W, to be only that due to overcoming the friction of the valves themselves through their whole travel, then if T be the travel of valves in inches; for a compound engine

$$W = \frac{T}{12}\left(\frac{L \times B \times p}{5}\right) + \frac{T}{12}\left(\frac{L^1 \times B^1 \times p^1}{5}\right), \text{ \&c.,}$$

and for an expansive engine with two cylinders

$$W = 2 \times \frac{T}{12}\left(\frac{L \times B \times p}{5}\right); \text{ or } \frac{T}{30}(L \times B \times p).$$

To provide for the friction of link-motion, eccentrics, and other gear, and for abnormal conditions of the same, take the work at one and a-half times the above amount.

To find the stress at any part of the gear having motion when reversing, divide the work so found by the space moved through by that part in feet, the quotient is the stress in pounds; and the size may be found from the ordinary rules of construction for any of the parts of the gear.

In modern high-revolution engines the weigh-shafts, levers, and all other parts of the reversing gear must be much more substantial than formerly to withstand the effect of inertia forces and to avoid excessive "give" and vibration when the engine is at full speed.

The Diameter of Weigh-shaft of compound engines may be fixed by the following rule:—

$$\text{Diameter} = \sqrt{\frac{\text{N.H.P.} \times \text{revolutions}}{F}}.$$

For two-crank engines F = 1,000, three-crank 900, four 800; very large ones may be hollow tubes.

N. E. Coast Inst. E. and S. rule is diameter weigh shaft not less than (diameter ot piston-rod — 1 inch).

CHAPTER XVII.

VALVE DIAGRAMS.

Motion of the Piston.—Fig. 156 illustrates the ingenious method introduced by Professor Zeuner for finding the position of the piston at any position of the crank, and should be used always when constructing a valve diagram, in order to find the points of cut-off, release, compression, &c.

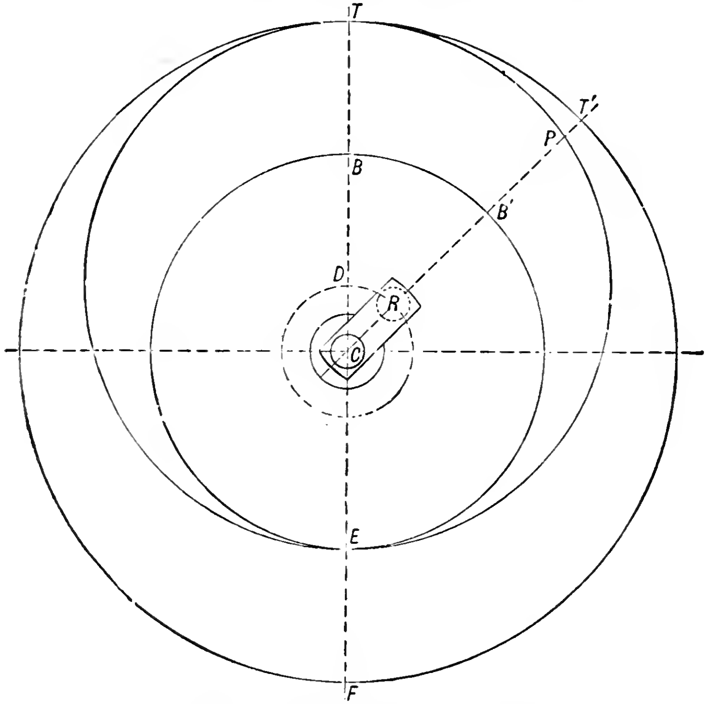


Fig. 156.—Diagram of the Piston Path.

The following is the construction of such a diagram :—Draw a line T F, and on it take a middle point C ; cut off a part C D, equal to the radius of the crank (to any convenient scale), and a part D T equal to the length of the connecting-rod.

With D as centre, and D T as radius, draw a circle cutting the initial line at E. With C as centre, and C T as radius, draw another circle ; and with C as centre, and C E as radius, draw a third circle. To find the position of the piston, with respect to its extreme position, for a position of crank C R when it has moved through an angle D C R from the "dead" point C D; produce C R to cut the circles at B′ P T′; then the piston has moved through a space T′ P in turning the crank through the angle D C R, and it is distant from the other end of its stroke by a space P B′. The correctness of this construction is easily seen by supposing an engine whose piston is connected directly to the crank-pin (such as the trunk engine) to be turned around about its shaft, while the crank remains stationary in the position C D. The circle T T′ F represents the path of the back end of the cylinder, and B B′ E that of the front end, since the cylinder is turned about the point C; the path of the piston will be on the circle T P E, since it is held at the same distance from the crank-pin D.

Diagram for the Common Slide-valves.—(1) Given the travel of the valve, amount of the lead or opening of valve at commencement of stroke, and the point of cut-off, to find the lap of the valve and the position of the eccentric, &c. These are the conditions generally predetermined in every-day practice.

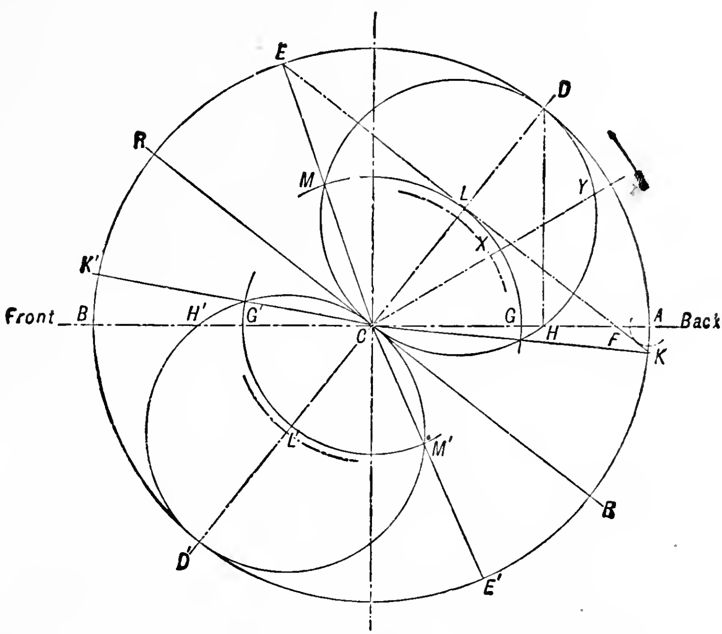


Fig. 157.—Zeuner's Diagram for the Common Valve Motion.

Fig. 157.—Draw a straight line, whose length A B is equal to the travel of the valve, and on it as diameter draw a circle, the centre of which is at C. Draw a line C E, so that A C E is the angle through which the crank has to move to arrive at the position of cut-off (this is to be obtained by drawing the piston diagram outside the circle A E B). With A as centre, and a radius A F equal to the *lead*, draw part of a circle, and through E draw a line touching the circle and cutting the original circle at K.

Through C draw a line C D perpendicular to E K, and cutting it at L. This line C D bisects the angle K C E.

C L is the amount of lap required, and B C D is the angle between the crank and eccentric to obtain the cut-off required; and since C D is the half-travel of the valve, L D is the maximum amount of opening of the port.

To extend the usefulness of the diagram, on C D as diameter, draw a circle, and with C as centre, and L as distance, draw the arc of a circle G L M, which is called the lap circle. The part G H intercepted by these circles, is equal to A F, and represents the *lead* or opening when the crank is in position C A; X Y likewise represents the opening of the port, when the crank is in position C Y.

To determine the operation of the valve at the other end, it is necessary to produce D C to D′ and on C D′ as diameter, draw a circle. Let H′G′ be the "lead" at the other end of the valve; with C as centre, and C G′ as radius, draw another lap circle, cutting the circle C D′ at M′. Through C M′ draw a line C E′. Then C E′ is the position of crank at cut-off, it having moved through the angle B C E′ and C G′ is the amount of lap required.

(2) If, however, the *travel*, *lap*, and *lead* are given to find the position of eccentric and cut-off, the following is the construction:—

Draw on A B as diameter, the circle as before. Cut off C G equal to the lap, and G H equal to the lead. At H, erect a perpendicular to A B, cutting the circle at D. Join C D, and on it as diameter, draw a circle; with C as centre, and C G as radius, draw the lap circle, cutting the circle C D at M. Draw a line through C M, cutting circle A D B at E.

Then C E is the position of cut-off, and B C D is the angle between the crank and eccentric.

(3) Given the *travel*, *lap*, and *position of eccentric*, to find the cut-off, lead, &c. Let B C D be the angle between the crank and the eccentric. Draw the travel circle as before, and on C D as diameter draw the circle cutting A B at H. Draw the lap circle G L M, then C M E is the position of cut-off, and G H is the lead.

The position of the crank when the valve commences to open is C K, or at the angle A C K, with its initial position.

If the valve has no inside lap—that is, the ports are both just closed to exhaust when the valve is in mid position—the position of release is at C R, a line at right angles to C D; and the position at which compression takes place is C R′, also at right angles to C D.

If, however, the valve has *inside* lap amounting to, say, C G′, release will not take place till the crank is at C K′.

If, on the other hand, the valve is cut away on the inside so as to have negative lap to the amount of C M, then release will take place at C E, and compression at C K.

If the inside lap is less than these amounts, the position of release and compression can be found by drawing an inside lap circle with a radius equal to the lap, and through the points where it cuts the circle on C D and C D′, the lines drawn through C and those points of intersection will give the positions of crank.

Travel of Valve.—It is usual to fix the travel of the valve before determining any further particulars, as so many things depending on this have often to be considered before there is leisure to finally decide the lead, lap, &c.

It will be seen, on reference to the diagram, that the opening to steam will vary with the travel, and if the *area* of opening is fixed from certain considerations before-mentioned, it is an easy matter to calculate how much the valve must open to give that area. Now the opening, together with the lap, is equal to half the travel, and with the same *positions* of "lead" and cut-off, the opening will be a constant ratio of the travel. This ratio can be determined by drawing a preliminary diagram with an assumed travel of valve. Let R be this ratio, and Q the amount of opening desired; then

$$\text{Travel of valve} = Q \div R.$$

Since it is usual to design the ports of a steam cylinder so that the flow of steam when exhausting may not exceed a certain velocity, it is evident that the port should open fully for that purpose; hence the travel of the valve, when there is no inside lap, should not be less than twice the length of the port, and is generally about $2\frac{1}{2}$ to 3 times the length of the ports.

The less travel the valve makes, the less work is absorbed in moving it, as the work is very nearly proportional to the travel. Double- and treble-ported valves are resorted to with the object of reducing the travel, as by them double and treble openings are obtained, and the travel may, therefore, be one-half and one-third that of the common valve with single ports.

"**Lead.**"—The amount of lead given to the valve is generally decided arbitrarily and according to judgment or prejudice. It will be seen by referring to fig. 157 that with the same "lap" an earlier cut-off is obtained by moving the eccentric farther from the crank, but at the same time increasing the "lead;" if the earlier cut-off is to be obtained without altering the "lead," the lap must be increased and the eccentric moved; but if the lap be increased, the opening is decreased, and the resistance at the port thereby increased. If, therefore, an early cut-off is required without the aid of an expansion valve, the "lead" must be great or the travel great, to get sufficient opening to steam, or the lead and travel must be both larger than would be the case with a later cut-off.

The considerations which should operate in deciding the amount of lead are the speed of the engine and the inertia of moving parts compared with the piston area. The momentum of the piston and rods should, if possible, be absorbed in compressing the steam remaining in the cylinder when the valve closes to exhaust; and were this always the case, there would be no need for fresh steam to enter the cylinder until the piston was at the end of its stroke; but this, under ordinary circumstances, seldom occurs, and if no fresh steam were admitted, "to form a cushion" for the piston, there would come a considerable jar on the bearings, etc., at the end of every stroke, owing to the strain increasing the displacement of the shaft in its bearings, and the sudden application of the load when the valve opens, causing its replacement with considerable force. This action is also observable when there is considerable "lead" without adequate "compression;" and although the "hammering" is usually attributed to excessive "lead," it is really due to want of adequate compression, for on "notching-up" the link, it generally ceases, notwithstanding that the lead is then thereby very considerably increased. For really quiet and sweet running there should be plenty of compression and no lead—even negative lead is of advantage at high rate of revolution, provided the valve eventually opens wide enough.

Long-stroke engines may have considerably more lead than those of the same cylinder capacity with shorter stroke, as the weight of moving parts is not less and sometimes more, the piston velocity is more (with the same number of revolutions), while *the piston area is less.* Fast-running engines without considerable compression may also have more "lead" than slow-running ones, as the element of time bears on the operation of the steam. Again, if owing to the valve motion the valve opens slowly, the period of admission may be earlier than should be the case with a quick opening one, as the wire-drawing at the commencement of admission will produce a similar effect to compression.

The amount of lead at each end will vary for two reasons; first, the cut-off at each end being effected by the same eccentric, any variation in position of cut-off must be obtained by varying the lap, and any variation in the lap will cause an inverse variation in the lead; secondly, the lead should be less at that end of the valve remote from the gear, as the adjustment after wear of the latter tends to increase the "lead" at that end. If the cut-off is equal at both ends the laps will vary, that at the end of a direct-acting engine remote from the shaft being more than at the other. With the ports as generally found in every-day practice, and with such travels as are practicable, the cut-off at the end near the shaft should be earlier than at the end remote from it, as the larger opening got by the decreased lap causes a fuller indicator-diagram at that end. For these reasons many engineers arrange so that the lead at the back or top end is only half that at the front or bottom end.

Inside Lap.—The effect of positive lap on the exhaust side of the valve is to retard the period of release, and to increase the compression by accelerating the closing to exhaust. The effect of negative lap is, of course, the reverse of this, and is also to give a full opening to exhaust sooner than would be the case with positive lap. A large amount of lap, either negative or positive, is required on the inside of a valve to materially change the periods of release and closing to exhaust, because just then the valve is moving at its quickest speed; but a small amount of negative lap makes a considerable difference to the amount of opening to exhaust at the end of the stroke, when the steam should exhaust wholly from the cylinder. There is no advantage in retaining steam in the cylinder to the very end of the stroke, as the effort of the piston on the crank is then ineffective to produce good results, and any forward pressure only serves to increase the difficulty of bringing the piston gently to rest by means of cushioning, etc., and this is especially true of quick-moving engines, whose steam ports are comparatively small.

If the valve has negative lap on the exhaust side, it is manifest that at a certain period there is a momentary communication between both ends of the cylinder; when the negative lap is considerable, and the ports small, the effect of such a communication is to fill the exhausted cylinder on one side of the piston with the steam released from the other side, just before the valve closes to exhaust, and is seen very distinctly on the indicator-diagrams, in the form of a sudden rise of pressure at the commencement of the compression. In cases of this kind, ample compensation is made for the retarding of the compression by negative lap in the increase of back pressure at the commencement of compression.

The high- and medium-pressure valves of compound screw engines should always have a considerable amount of negative inside lap, to allow of a continuous and easy flow of steam during exhaust. The amount of negative lap on the inside should be *always* less than the outside lap of the valve, otherwise there will not be enough bar to cover the port, so that steam can pass from the valve-box to the exhaust just as the valve is opening and shutting; for a quick-running compound engine there should be such lap that exhaust commences from the high- and medium-pressure cylinder at 0·9 of the stroke; and with a slow-working paddle engine not later than 0·95 the stroke.

Effect of "Notching Up."—When an engine is fitted with link-motion or other means of reversing it without throwing "out of gear," so that the mechanism which would produce stern motion, can be made to effect that producing ahead motion, a certain variation in the cut-off, release, &c., can be effected by what is called "notching up." The origin of this term is clearly traceable to the locomotive, whose reversing lever is held in place by a sliding-bolt fitting into a *notch* in a quadrant provided for the purpose.

When the link is notched up so that the block works the valve from a point between its two extreme positions on the link, the motion is one due to the combined effort of both eccentrics, and in order to examine clearly the operation of the valve under these circumstances, it is necessary to find the position and throw of the *equivalent eccentric.* To determine this exactly is somewhat difficult, but a very close approximation can be made by the method suggested by the late J. Macfarlane Gray, as follows:—

Suppose the link (fig. 159) to be notched up to a point M; or, in other words, the link moved so that the link block is distant M T from the point at which the eccentric-rod is attached to the link.

Draw the valve diagram (fig. 158) due to the position and throw of the eccentrics of fig. 159; and through D D′ draw the arc of a circle with a radius found as follows:—

Radius = length of eccentric-rod from centre to centre × half the distance between the centres of the two eccentric sheaves ÷ the distance between the centres of eccentric-rod pins on the link.

$$\text{Referred to fig. 159,}\quad \text{Radius} = \frac{\text{D T} \times \text{D D}'}{2\ \text{T N}}.$$

$$\text{On this arc, take a point Z, so that } \frac{\text{D Z}}{\text{D D}'} = \frac{\text{T M}}{\text{T N}}.$$

Join C Z, and on it as diameter draw a circle, cutting the lap circle at points L and E; through O L and O E draw lines which represent the position of opening and closing respectively at one end of the valve when the link is notched up to the point M.

It will be seen that the effect of notching up is the same as would be produced by an eccentric whose position on the shaft is at the angle B C Z with the crank, and its eccentricity C Z; that the opening to lead is earlier, and the magnitude of the lead considerably increased; that the reduced travel gives a smaller opening to steam, and that the cut-off is earlier; likewise, by examining further, it can be seen that release takes place sooner and compression commences earlier than when in full gear.

When the eccentric-rods are arranged as shown in fig. 159, they are said to be *open*, or it is an *open rod link-motion;* if, on the other hand, D N and

D′ T be joined, it is said to be a *crossed rod link-motion*. When the link motion has crossed rods, the arc D Z D′ should be convex to the point O, that is, the centre of curvature is on the side A.

When the rods are crossed, an earlier cut-off is effected by notching up without materially altering the *lead*, and, in most cases, with a very early cut-off there is only a very small *lead*, smaller even than when in full gear. But the travel is reduced very rapidly by notching up, giving a corresponding reduction of opening to both steam and exhaust; so that, altogether, crossed rods are not so convenient as open ones for working expansively by notching up the link, and are very seldom so fitted.

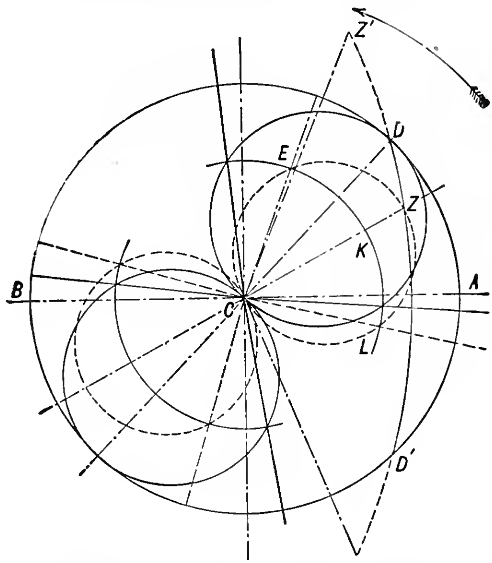


Fig. 158.—Diagram showing the Effect of "Notching up."

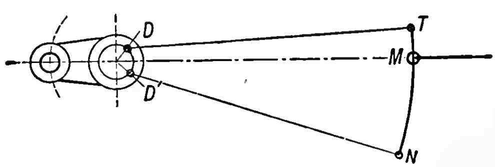


Fig. 159.—Link Motion "Notched up."

The valve diagrams, when the link-motion is such as shown in fig. 159, are constructed on the same principle as for notching up, and the position of the eccentrics and their throw are determined by producing the arc D D′ (fig. 158), and taking a point Z′ *beyond* D, so that

$$\frac{Z' D}{D D' + 2 Z' D} = \frac{T M}{T N}.$$

Join C Z′.

Then C Z′ is the eccentricity of the sheaves, and B C Z′ is the angle between them and the crank.

Expansion Valve on an Independent Face, Central Position Ports Closed.—The valve in this case is working under precisely the same conditions as a common slide-valve, and the same construction of diagram as fig. 157 holds good.

Expansion Valve on an Independent Face, Central Position Ports Open.—Let C O′ (fig. 160) be the position at opening of valve when set at earliest cut-off, which should be somewhat before that of the main valve, and C E the position at the earliest cut-off which is required. Bisect the angle O′C E by the line C T, and make C T equal to half the maximum travel of the valve.

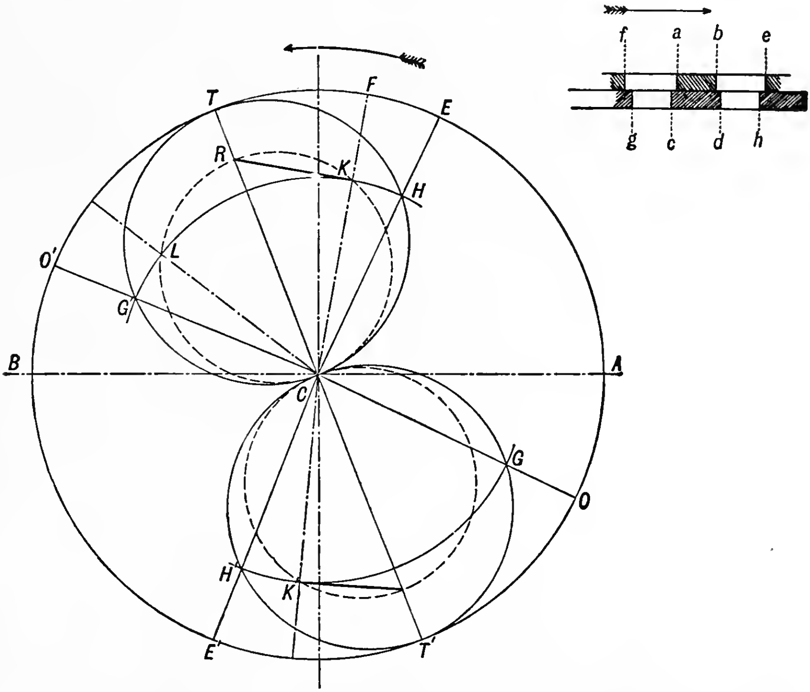


Fig. 160.—Diagram for an Independent Expansion Valve.

On C T as diameter, draw a circle cutting C E at H.

Produce C T to T′, making C T′ = C T.

On C T′ as diameter, draw a circle cutting C O at G.

Then C H is the distance of the cutting-off edge of the valve from the edge of the steam port; that is, C H = $b\,h$; and the lap of valve $b\,d$ = C H − length of port.

The width of the bar $a\,b$ must not be less than C T − $b\,d$, and should be = (C T − $b\,d$) + ¼ inch.

For the other end of the valve, a similar construction will give the required results.

Let C E′ be the position of cut-off at the other end of the cylinder, cutting the circle C T′ at H′.

With C as centre, and C H′ radius, draw a circle cutting the circle C T′ at G; through C G draw a line, then C G is the position at which the valve opens again.

Hence $a\,g$ = C H′; and the lap $a\,c$ = C H′ − length of port.

Again, the width of bar $a\,b$ must not be less also than C T − $a\,c$; and should be = (C T − $a\,c$) + ⅛ inch.

The bar $c\,d$ = $a\,b$ + lap $b\,d$ + lap $a\,c$.

To find the reduction in travel for a later cut-off at position C F cutting the port circle at K. Draw K R perpendicular to C F cutting C T at R. On C R as diameter, draw a circle cutting the lap circle at L. C R is the half travel of valve. C L is now the position at which the valve opens again, which should not be before the main valve is closed to steam.

In arranging an eccentric and gear for this valve, care must be taken in setting it that the expansion valve will open at the earliest cut-off before the main valve opens, and that at the latest cut-off it does not open again till the main valve is closed. When the expansion valve spindle is in the same plane with the piston-rod, and the gear is direct from the eccentric to the valve-rod, the eccentric should be set at the angle B C T with the crank on the same side as the *head-going* eccentric.

Expansion Valve Working on the Back of the Main Valve, Cutting off at the inside edge, and Variation in Cut-off made by Varying the Travel.—Let A (fig. 161) be the dead point of the crank at the back of the stroke, and C E the position of earliest cut-off. B C D is the angle between the crank and *head-going* eccentric, and C D its eccentricity. Cut off C H equal to the lap of the expansion valve, or distance of the inner edge of expansion valve from inner edge of steam port in the back of the main valve, when the expansion valve is in its mean position with respect to the main valve. Draw the lap circle H G as before. From H draw a line H O perpendicular to C E with D as centre and the half travel (absolute) of the expansion valve as radius, draw the arc of a circle cutting H O at O. Join D O and C O; on C O as diameter, draw a circle, which will pass through H, since C H O is a right angle. Complete the parallelogram C D by drawing D T parallel to O C, and C T parallel to D O. Then C O is the half-travel of the expansion valve relative to the main valve; B C T is the angle between the crank and the expansion eccentric, or T C D is the angle between the expansion eccentric and the head-going eccentric.

By producing O C to O′, and on C O′ drawing a circle, &c., the cut-off, lap, &c., may be investigated for the other end of the valve.

To find the effect of shortening the absolute half-travel to C R. Since the position of the eccentric remains unchanged, join R D, and through C draw C N parallel to R D, and cutting O D in N. On C N as diameter draw a circle cutting the lap circle at L and K.

Through C L and C K draw lines which are respectively the positions of opening and cut-off of the expansion valve for the back end, and C N is the relative half-travel.

It will be seen that when the travel of the expansion valve is nothing, that the relative half-travel is C D, and that if the lap is the same as that of the main valve, the cut-off and lead will be the same also. For this reason it is customary when designing a valve of this kind, which may have to go out of gear when at full speed, or for some particular purpose, to make the lap the same as that of the main valve, and for practical reasons to make the travel the same also.

To find the effect of the expansion valve when in stern-gear, join T to D′ of the stern-going eccentric, and complete the parallelogram C D′ as before.

It will be seen also, that with this kind of expansion valve, the opening to steam is never less at the expansion valve ports than at the cylinder ports.

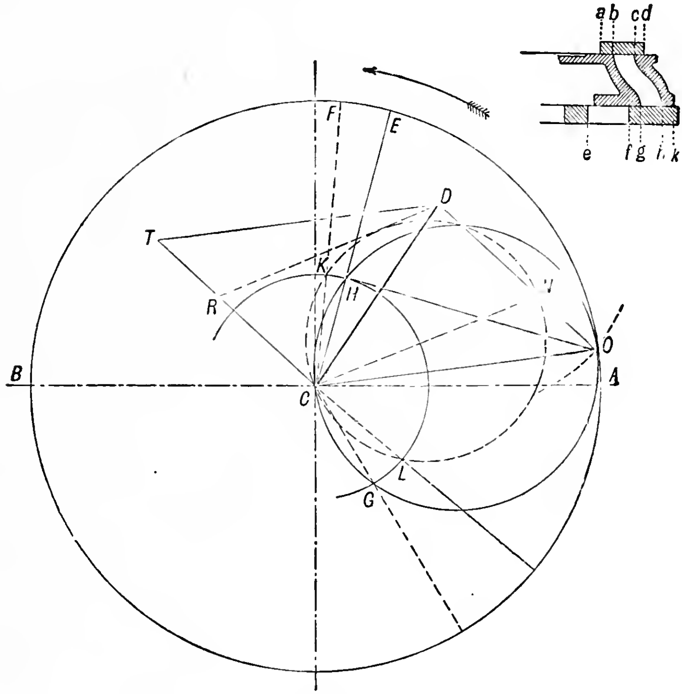


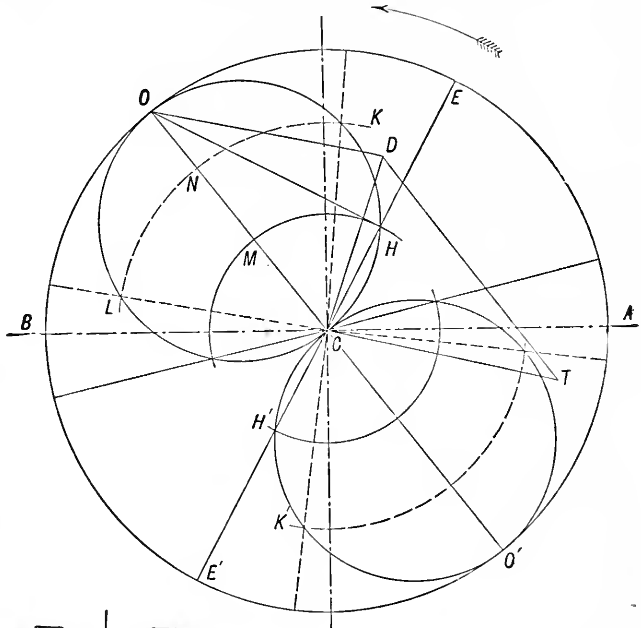
Fig. 161.—Diagram for an Expansion Valve.

Expansion Valve Working on Back of Main Valve, and Cutting off at Outside Edge—Let A (fig. 162) be the dead point of the crank, as before, at back of stroke, and C E the position of earliest cut-off required, B C D is the angle between the crank and the head-going eccentric, and C D its eccentricity. From C E cut off a part, C H, equal to the opening to steam of the main valve at that end, or such as would give sufficient opening by the rules as laid down already. From H draw a line, H O, perpendicular to C E. With D as centre, and D O a radius equal to the eccentricity of the expansion eccentric, draw the arc of a circle cutting H O at O. Join C O, and on it as diameter draw a circle, which will pass through H, since C H O is a right angle.

Complete the parallelogram C D by drawing D T parallel to C O, and C T parallel to O D.

Then C O is the half-travel of the expansion valve with respect to the main valve; B C T is the angle between the crank and the expansion eccentric, and T C D the angle between the expansion eccentric and the head-going one of the main valve. C H is the distance of the edge of the expansion valve (fig. 163) from the outer or cutting-off edge of the port on the back of the main valve, when the expansion valve is in its mean position with respect to the main valve.

If the expansion eccentric is set *exactly* opposite to the crank, so that the point T is on the line A B, then the cut-off is the same in *stern-going* as in *head-going* gear. If, however, it is set as shown in fig. 162 so as to be nearer the stern-going eccentric, the cut-off is later in *stern-going* than in *head-going gear*, and the relative travel of the expansion valve is greater.



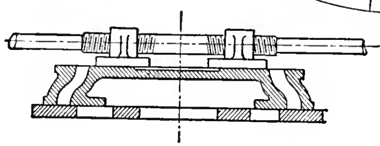
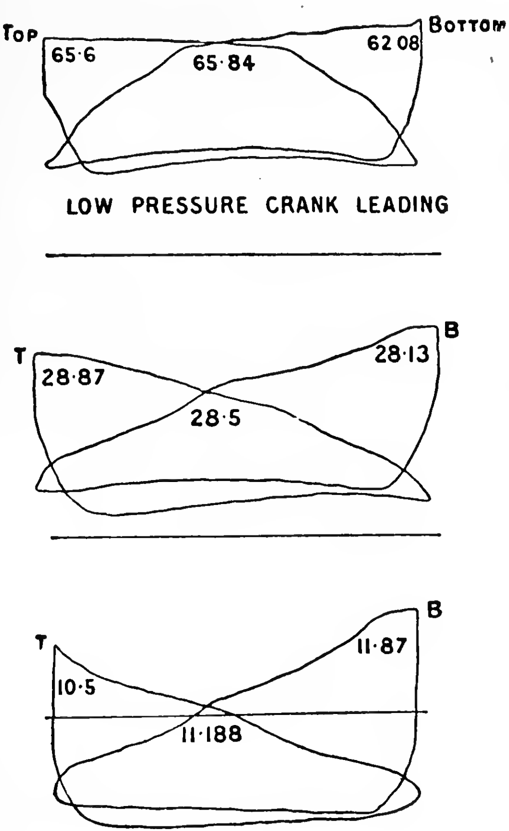


Fig. 163.—Expansion Valve. Fig. 162.—Diagram for an Expansion Valve.

If it be required to cut-off at a later period, as at C K, the expansion valve must be so moved that when in mean position with respect to the main valve, the cutting-off edges are apart by a distance equal to C K; that is, the valve is moved towards the middle by a distance M N or C K − C H.

The laps, cut-off, &c., at the other end of the valve may be investigated in a similar way by producing O C to O′, making C O′ equal to C O, and on it as diameter drawing a circle cutting the required position of earliest cut-off O E′ at H′.



S.S. "Flamborough."

Cylinders 22, 35, 58 ins. diameter.
Stroke 42 ins.

	Low P. Leading.	High P. Leading.
Boiler press., lbs.	150	150
Vacuum, . ins.	26	26
Intmdt. recvr. press., . lbs.	54	58
Low recvr. press., . lbs.	8	7
Revolutions, .	67	62
Mean press., H.P.	63·8	65·9
,, M.P.	28·5	32·8
,, L.P.	11·19	10·12
,, referred,	30·73	31·55
I.H.P., . H.P.	344·8	329·4
,, . M.P.	389·6	415·1
,, . L.P.	420·1	351·8
,, . Total,	1154·5	1096·3

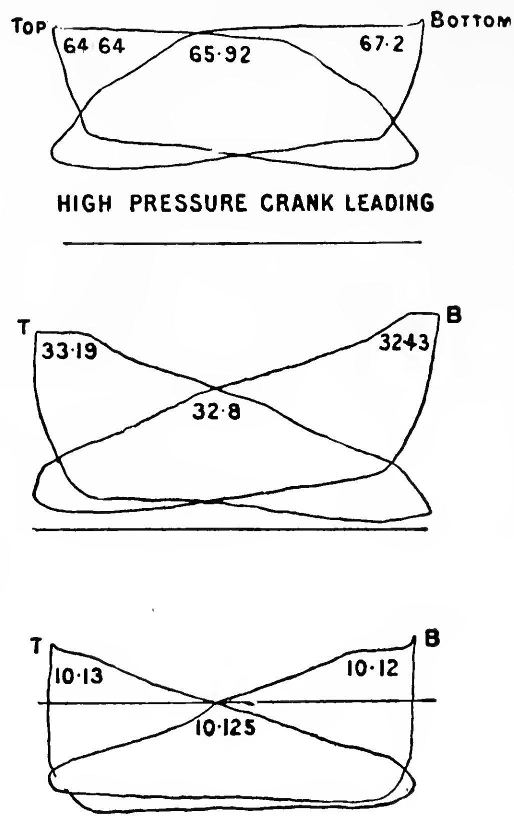


Fig. 164.—Effect of Crank Sequence.

C H′ is then the distance apart of the cutting-off edges of the valves at the other end. Also, if a later cut-off C K′ is required when C K is the cut-off at the other end, the valve must be moved towards the middle by a space equal to C K′ - C H′.

If C K be the position of latest cut-off, care must be taken that the arrangement is so designed that the re-opening of the port at C L is *after* the main valve has cut off to steam.

In order to obtain as large a relative travel as possible to the expansion valve, it is customary to make the angle B C D about 100°, so that the main valve has of necessity very little lap and small lead, and consequently there is a late release and a very small amount of compression.

In speaking of C D as the eccentricity, and B C D the angle of the head-going eccentric, it must be understood that an eccentric acting directly and in line with the main valve spindle is meant. If the valve gearing is such that the eccentric is otherwise, then C D is the eccentricity and B C D is the angle of the *equivalent eccentric.*

This remark applies likewise to the case of the expansion valve (fig. 163) cutting-off at the inside edge.

Construction of Valve Diagrams.—In making the diagram for an engine, the piston diagram (fig. 156) should be drawn first to such a scale as to be well clear of the valve diagrams. The main-valve diagram should then be drawn full size in black ink, and the expansion-valve diagram on it in red, or other distinctive colour. The effect of each valve can then be traced both separately and conjointly.

Effect on Indicator Diagrams of Crank Sequence.—Fig. 164 shows the effect of high-pressure crank leading the M.P. crank, and also the effect of the low-pressure crank leading the mean-pressure. It will be seen that in the former case the loads on the pistons are greater at the beginning of the stroke than is the case when the low-pressure leads. This is a good thing, especially in fast-running engines where the inertia of the moving parts is considerable; the ratio of maximum to mean load is greater in this case, and the variation in temperature is greater than when the low-pressure leads.

CHAPTER XVIII.

PROPELLERS.

The Fundamental Principle on which all Marine Propellers act is that the reaction from a stream of water projected from a floating body is equal to its action in magnitude but opposite to it in direction; and that reacting force called *thrust* applied to the body tends to produce motion of it in that opposite direction.

The stream of water may be produced by any instrument either within the ship, as by some form of pump, or without as by a paddle-wheel or screw. The object of the engineer is, therefore, to obtain in any case as large a thrust as possible with the least amount of water disturbance, and the efficiency of the means he employs to propel a ship is measured by such a comparison, whereas the maker of pumps aims at disturbing as much water as possible with the least amount of thrust, inasmuch as to produce the thrust he has to expend energy just as the marine engineer has in disturbing the water; the end and the means being transposed, the aim is therefore different, and in considering propeller problems it is desirable to free the mind from too much analogy to pumps otherwise grave misconceptions may arise which will lead to fallacious conclusions. It may also be impressed on the mind, while dealing with the question of efficiency of propeller, that mere thrust in the gross is not the true measure of the success of a propeller, for a screw may, and often does, create a serious amount of augmented resistance, so that its actual or net propelling effect is less than another screw which does not produce nearly so much gross thrust. In former times, when the tendency was to supply every steamship with a propeller as large as her draft of water and construction permitted, and to make the surface and breadth of blade proportional to the diameter, it was no uncommon thing to find the propulsive efficiency very poor notwithstanding exceedingly high thrusts. The phenomenon of negative slip generally showed itself as an accompaniment to such screws; and although the so-called negative slip is not absolutely incompatible with good results, satisfactory ones seldom do accompany it.

If a steamship is at rest with respect to the water in which it floats, and is to be set in motion by the stream of water projected from it, the inertia must be overcome, and until motion actually ensues the velocity of the stream is the same with respect to the still water as that with which it flows. Then the energy is for a time used in accelerating the motion of the ship, and with the increase in its motion there is a corresponding decrease in the flow relative to surrounding water. Moreover, as the ship increases in speed her resistance increases as the square of the speed, and following that law the increments lessen gradually until the maximum speed is reached, when the whole of the effective propelling effort is absorbed in overcoming the resistance—the net thrust is then equal to the gross resistance.

If V is the velocity of the issuing stream from the ship, and v is the velocity of the ship itself, both in feet per second, the velocity of the stream with respect to still water is $(V - v)$. This is called the *slip*, and to distinguish it from other features of marine propulsion, it is named *real slip*. It is generally expressed as a percentage of the velocity of flow, hence—

$$\text{Real slip per cent.} = \left(\frac{V - v}{V}\right) 100.$$

The propelling effect of a stream of water depends on its mass and velocity; the mass depends on its quantity, and that quantity on the sectional area of the stream and its velocity. If, therefore, M is the mass of water projected with a section of A square feet at a velocity V from a ship moving at a velocity v, and taking the weight of a cubic foot of sea water at 64 lbs., and the effect of gravity at sea level as 32—

The momentum of the stream $= M (V - v)$.

$$M = W \div g ; \quad W = A \times V \times 64.$$

Then, momentum of stream $= \dfrac{64 \times A \times V}{32} (V - v) = 2 A \times V (V - v)$.

The thrust or propelling force due to this stream is, therefore,

$$2A \times V (V - v).$$

All attempts at what may be called propulsion by internal propellers have proved failures in practice, however efficient the instruments themselves may have been. The losses from friction in the pipes, orifices, etc., the great inconvenience experienced by the presence of pipes, channels, etc., within the ship, in addition to the incumberance of the propeller itself, form an impassable bar to the use of such apparatus in any but small ships of quite low speeds, whatever the efficiency as propellers may be.

The Stream Water in Every-day Practice proceeds now from some form of propeller without the ship, and it is fed by a flow to it from the surrounding water mostly acting under the influence of gravity. With a screw deeply immersed, and well away from the body of the ship, the supply of water is ample, uniform, and free from air bubbles; it is, however, not free altogether from air, inasmuch as there is always a considerable volume of it held by the water in solution, otherwise fishes could not remain alive in it; if the propeller, from any cause, demands such a supply as to produce decrease in pressure at the back of the blades to be considerable, some of this air is liberated from solution, and appears as bubbles mechanically mixed with the water, and following the stream to the front or active face of the screw, it thereby materially reduces the thrust. This phenomenon is known as cavitation, from the fact that the action of propellers moving at high velocities with insufficient blade area is to make cavities in the stream flowing through them in which the freed air escapes and remains enveloped with water as bubbles. These must not be confounded with the air bubbles formed by the suck of the screw from above, producing air eddies which act as air "spouts," nor with those formed by the blade tips breaking the surface when there is not sufficient immersion. In both these cases the air mixes at once with the feed stream, producing disastrous results with the thrust, so that it may even at times disappear altogether for a moment.

When any Submerged Screw is working it is drawing to itself what was previously still water, and discharging in rear of it a mass of water which has to come to rest again sooner or later by the resistance of the surrounding water. It is certain now that the flow from it is a column nearly cylindrical, and that in the dispersion it retains the same form, although of a somewhat larger diameter than that of the screw. There is no circumferential dispersion, and so no need, in fact, for any means of retention, such as so many well-meaning inventors have patented "to save the loss due to centrifugal force action." The otherwise obnoxious water bubbles tell too truly the course of flow through and from the screw for any mistake to be made on that score. Prof. Flamm and Sir Charles Parsons have shown by exhaustive experiments and photography exactly what takes place, and having seen the performance of their model screws, it is easy to confirm their conclusions by making careful observations on real working screws in smooth, clear water.

In addition to the "head" of water itself, there is the further equivalent "head" due to atmospheric pressure (equal to 33·7 feet of water) imposed on the screw race to keep up a supply to the propeller to permit it to work without cavitation ensuing; but this latter pressure is also the cause of such a ready and plentiful supply of air to it when an eddy forms. It also conduces to the formation of the cavity or depression of the water surface just over the screw, into which the water in advance is always flowing under the influence of gravity as the ship proceeds. Abaft the screw there is always the tendency to pile the water up as the easiest way to disperse the column, the resistance being less near the surface than lower down; hence the wave or lump of water so often seen in the wake of a screw.

The Effects of Wake Currents on a Screw, particularly on the central one, is considerable, especially in ships of high speed with rather full "buttock" lines. In a sailing ship—that is, one without a screw—the flow of water into the void made by the ship is not limited to the streams following its lines or form; and when the speed is increased the tendency for such streams from either side to meet abaft the rudder is great; the space left by the ship must be filled, therefore, either from below or by water flowing in from behind under the action of gravity. The latter is the easier method, and the one usually observed. If a screw of considerable diameter be set to work at the end of the ship, it requires to be fed with abundance of water, and so it causes a most distant suck on either side of the stern in front of the screw; gravity again forces the flow along the skin to be accelerated, and induces a further supply to come from abreast. The shape of the ship at the stern causes the stream to follow a certain course, after which it and the screw's action divert it outwards again, so that the axial flow through the screw is less than that of the undisturbed water of the sailing ship—in fact, with respect to still water, the feed has forward motion with the ship. Inasmuch, then, as every screw is working in more or less disturbed water, so that no one can possibly say for certain what is the real value of V, the real slip is a matter of conjecture, but a fairly close approximation can now be made to it, thanks to the investigations and experiments of Dr. R. E. Froude, Prof. Taylor, and other patient and pertinacious men of genius and opportunity.

With the Paddle-wheel the Problem of Propulsion is simpler, inasmuch as the side wheel is working in water comparatively undisturbed, it produces but little augmented resistance, and the acting surfaces are generally normal to their action. On the other hand, in its course it breaks some of the water into foam, so that the surface stream has not nearly the density of still water, and a considerable portion is at times dispersed into the air. When, however, the ship is fairly underway the paddle-wheel is much more efficient than it appears, as the amount of foam and broken water then is more apparent than real. There is, however, the same tendency to formation of hollow due to sluggishness of feed, and there is the piling up of water abaft, which is inevitable, as there is only a small amount of side dispersal—in fact, the stream is rectangular from the wheel to its race end more or less, as that of the screw is cylindrical—and it is quite as difficult to estimate the real slip of a paddle-wheel as it is of a screw, and so the practical engineer has to be content to take the nominal slip.

The Nominal or Apparent Slip of a Propeller is what the slip should be if there were no interference, and the propeller worked as by theory it should do. The velocity of the stream is assumed to be that calculated by multiplying the lineal effect of the propeller by the number of revolutions. In the case of a paddle-wheel, the circumference at the centre of effort of the floats is taken, and with a screw the *pitch* or axial distance apart of two threads of one convolution.

If D is the diameter of the circle through the float effective centres, which is generally now taken as the centres on which the floats oscillate in a feathering wheel; P is the pitch of a screw, and R the revolutions per second of either; then in case of

(1) The paddle-wheel, $V = \pi D \times R$ feet.

$$\text{Apparent slip per cent.} = \left(\frac{V - v}{V}\right) 100 = 100 - \frac{31 \cdot 9\ V}{D \times R}.$$

(2) The screw, $V = P \times R$ feet.

$$\text{Apparent slip per cent.} = \left(\frac{V - v}{V}\right) 100 = 100 - \frac{100\ v}{P \times R}.$$

The Actual or Real Slip must be in every ship in excess of the nominal or apparent. In some ships the excess is great, not only because the wake currents are great, but because the stream of water projected is not really a solid cylindrical column. It is, of course, highly desirable that the stream shall flow at an uniform rate throughout its transverse section—that is, there shall be no difference in the rate of flow from the tip than from that at the middle or root of the blade—there should be no envelopes or cylinders of water passing through or sliding over one another. In practice such a consummation is scarcely possible, hence the real slip probably varies very considerably from root to tip, and there will be always a core following the boss or hub with no imparted velocity whatever.

The Spiral Path of the Blade Tips, as of every other portion of the blade, does no doubt by skin friction cause to be induced a more or less spiral flow to the wake; and probably as the feed approaches the screw it begins to assume a somewhat spiral path. Hence the column flowing from the screw is more like a common wire rope than a solid bar; so that if there are differences of flow and slip there will be layers or cylinders revolving as well as sliding in one another. The screw that avoids this action is a desideratum.

The Selection of Propeller Dimensions and Form is, from these disturbing causes, rather a matter of experiment and determination by empiric formula derived from successful practice, than direct from mere considerations of first principles and theory, involving more or less complicated mathematical calculations, as well as some necessary assumptions. Dr. R. E. Froude and Professor Taylor have, however, done so much to clear the problems of complications, and by exhaustive experiments produced data of inestimable benefit, that a good propeller can be designed which shall suit the conditions under which it has to work, and permit of at least good efficiency of its motor; further, by their help, the best propeller neglecting such limitations can be indicated. But, after all, it is only by actual trial on the real ship at sea under average working conditions of weather that one can say for certain that the screw is quite satisfactory. It is, of course, the prime duty of the designer to provide a screw that will work well and with the highest efficiency *on service and under service conditions*, rather than one which on trial in smooth water will produce the highest speed for a limited period under mere trial conditions, in spite of the ever-present temptation to this latter course. It is not an uncommon thing nowadays to hear that turbine-driven steamers on service have not fulfilled the expectations of their promoters, or done so well as on the trial trips. Such things have happened frequently in the past, when the trial trip has been the crucial test of the ship's performance in the minds of those who determined the proportions of the screws. Even turbine steamers may suffer no actual loss on service if the screw is of somewhat coarser pitch than requisite for the speed of the turbine itself to produce maximum joint efficiency.

The Common Paddle-wheel with Fixed Floats is now seldom seen*; it was the oldest and simplest form of marine propeller, and consisted of a framework of wrought-iron bars, braced together and stayed to prevent racking, attached to a cast-iron hub or boss keyed (though originally staked) to the shaft end. The boards or floats were secured to the radial arms just inside the outer rim by means of bolts having a hook end, which fitted on the arm. No doubt it was a development of the "under shot" mill wheel of the old millwrights. It was very simple in construction, strong and easily repaired if damaged, and replaced if worn out. For river craft carrying only passengers and parcels with little variation of draft of water, the radial wheel was fairly efficient; owing to the comparative ease with which the floats could be shifted or "reefed," it had advantages on a sea-going craft making fairly long voyages, for, in spite of a somewhat reduced efficiency under the circumstances as the "dip" of the float was reduced thereby, the engine was permitted to make the full number of revolutions for maximum power. These wheels to-day are used only on small, light draft, river craft in tropical regions, where a feathering wheel would be liable to derangement, and with no means for effecting repairs to it.

The Feathering Float Paddle-wheel, introduced by Elijah Galloway into general use in 1830, has continued to the present time as the accepted and best form of paddle propeller. The general idea of such a contrivance is

* It is, however, still in general use on American rivers, and preferred by U.S. engineers as permitting of a timber construction to a large extent, and consequently easy to repair *en route*.

to cause the floats to assume a position whereby they shall enter the water without shock and undue disturbance; to act in the most effective way during the passage through the wheel race; and finally to emerge without disturbance of water with as little resistance to the wheel torque as possible.

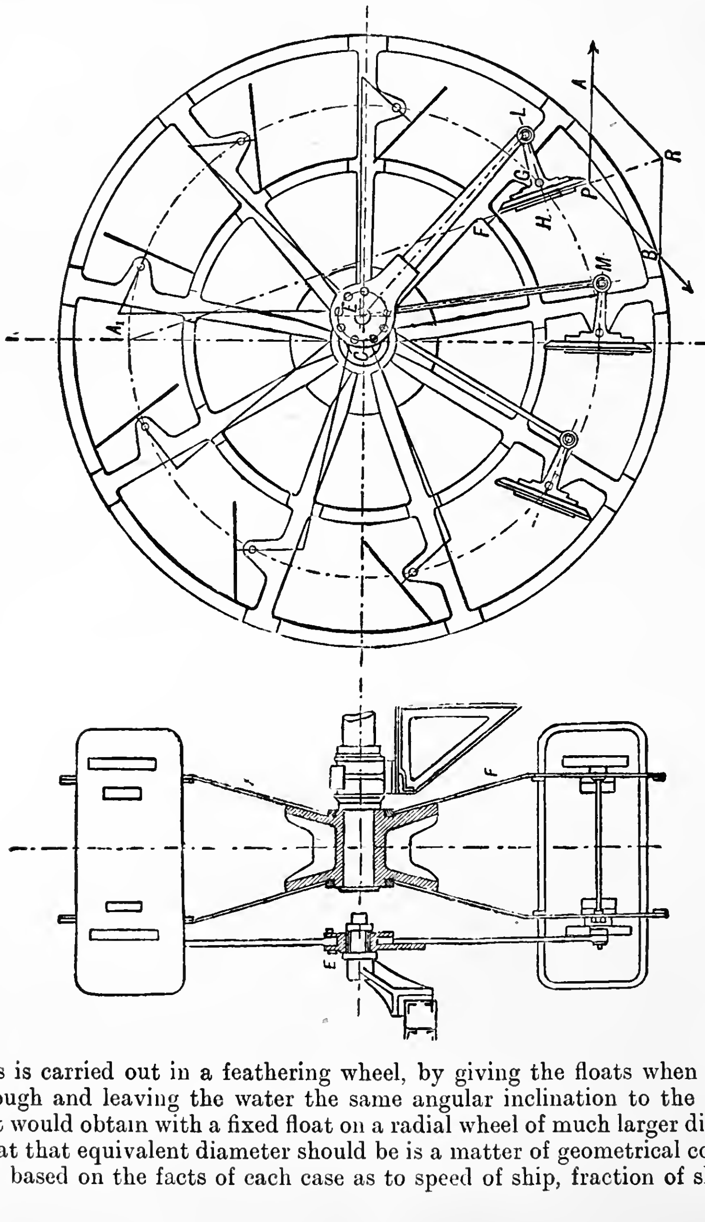


Fig. 165.—Paddle-wheel with Feathering Floats.

This is carried out in a feathering wheel, by giving the floats when passing through and leaving the water the same angular inclination to the vertical that would obtain with a fixed float on a radial wheel of much larger diameter. What that equivalent diameter should be is a matter of geometrical construction based on the facts of each case as to speed of ship, fraction of slip, and

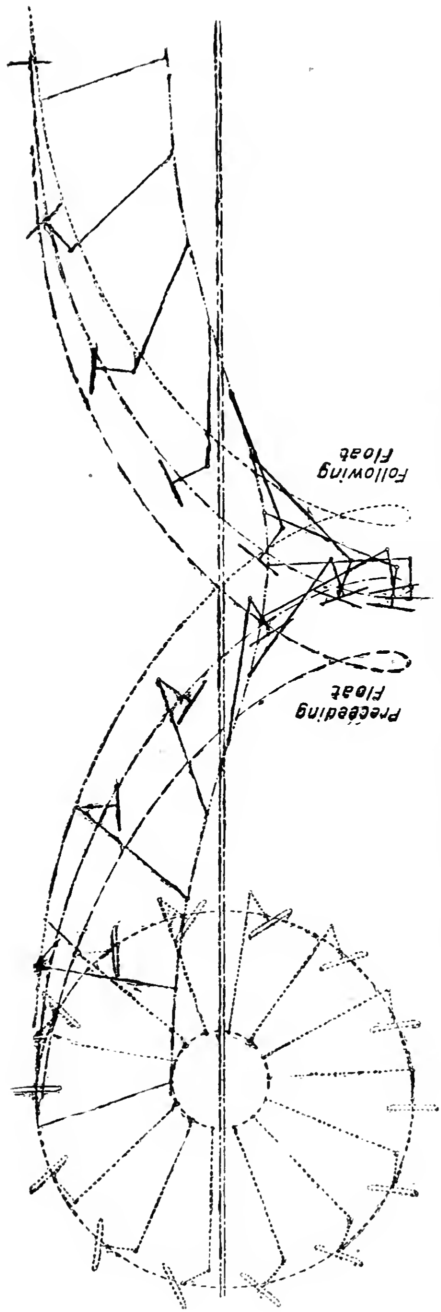


Fig. 166.—Locus of Floats and Float Centres of a Feathering Paddle-wheel.

actual diameter of wheel. To ascertain the proper position of a float at entry and the equivalent diameter of the wheel, take a point P (fig. 165) at the edge of a float just touching the water level. Draw line P A as representing the water level, and cut a part P A to represent the speed of the ship on a convenient scale; let P B be the tangent to the circle passing through P, and make P B as the speed of the wheel on the same scale. Complete the parallelogram A P B, whose diagonal P R represents the direction and rate of velocity of the point P at that particular moment of its course, and if the float face lies on the line made by continuing P R to A, it will start to enter the water edgeways with least resistance, and A_1 is the centre of the equivalent radial wheel. If the wheel is to work with considerable slip, the speed, as measured by P B, will be large compared with P A, this will mean an increase in value to A R, and a reduction in the angle made by A_1 R with the vertical and the raising of A_1. In other words, with greater slip the equivalent wheel must be greater. Fig. 166 shows the locus of the floats and float centres of a feathering wheel.

The Means provided for constraining the Floats are quite simple, and consist essentially of a fixed gudgeon, or its equivalent, eccentric to the wheel axis, its centre being in front of the shaft centre. On it revolves a ring, which receives its motion by an arm fixed to it and jointed to one of the float levers, so that it turns with the wheel; connected to it is a rod from each float lever, so that as the wheel turns round the eccentricity of the gudgeon causes the lever to vibrate through the angle necessary for efficiency.

The position of the gudgeon may be ascertained by placing a float with its lever in the proper position for immersion and emersion; with the lever pin centres as centre, and the radius of the pitch circle as radius, two intersecting circles can be described; then the point of upper intersection is the position of the centre of the feathering gudgeon. Sometimes, however, from practical considerations, the levers are at a less angle than 90° with the float face (*v.* fig. 167); then the length of the describing radii is less than that of the distance of float centres from wheel centre. Formerly with the slower-running engines an equivalent diameter twice that of the actual was common practice and sufficient. Nowadays, with the smaller wheels and the higher percentage of slip to give greater revolution rate, the equivalent diameter is much greater, and may be even three times, as shown in fig. 167.

The Diameter of a Feathering Paddle-wheel depends on the speed of the ship, the revolution of the engines, and the amount of slip. It is, of course, evident that if, for any reason, the sectional area of the projected stream of water from a propeller must be restricted or reduced there must be an increase in the rate of flow, so that the effect may be the same. It has been already shown that when $(V - v)$ is the slip—

$$\text{Thrust or momentum of water} = 2A \times V\,(V - v)\ \text{pounds.}$$

The increase in $(V - v)$ can be made only by an increase in V, ina much as the speed of the ship v is fixed

This means that the float velocity must increase as the reduction of area takes place, but A will vary inversely as V^2. With the modern paddle steamer of high speed it is, of course, desirable and even absolutely necessary to have the floats, and consequently the wheel, as small as possible; on the other hand, in order that the machinery may be of minimum weight and

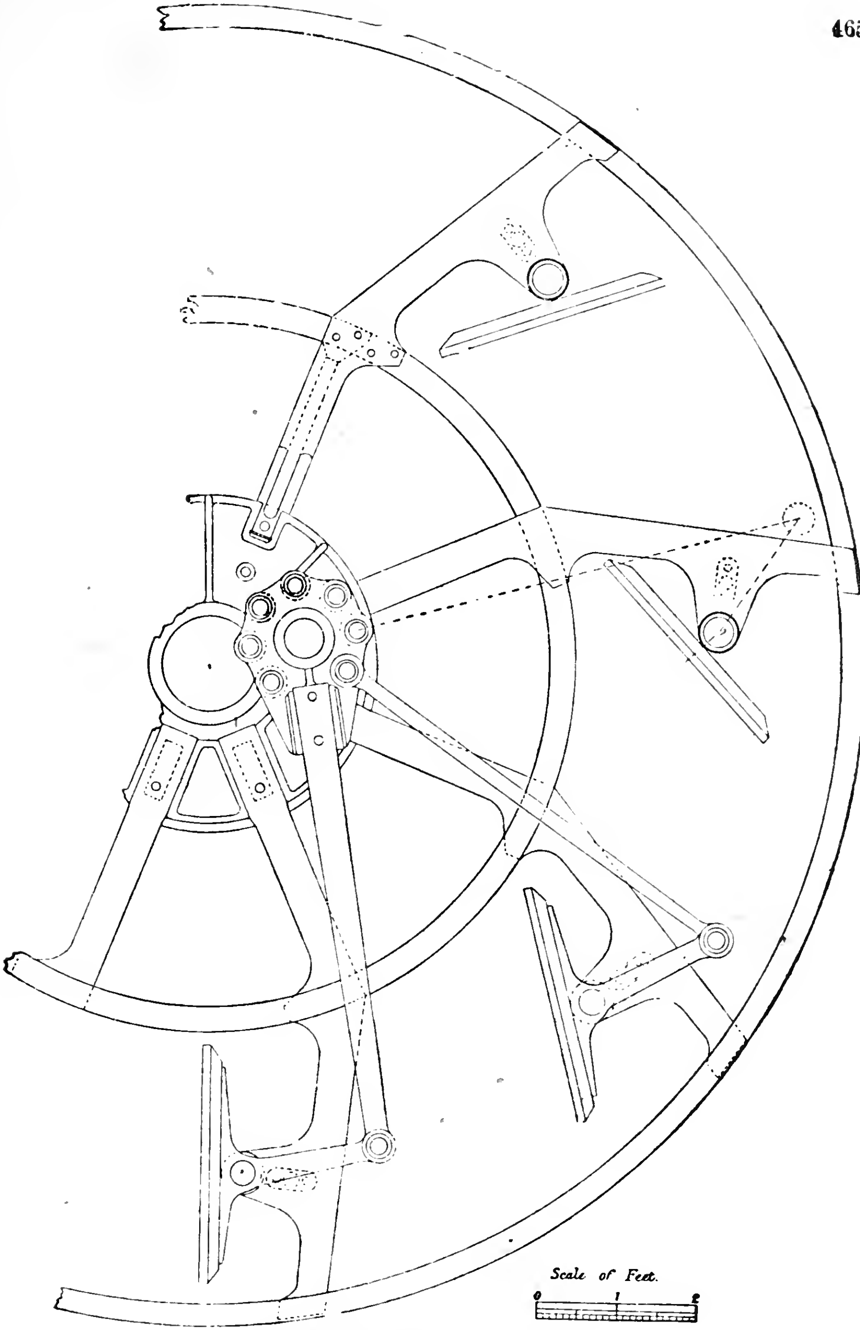


Fig. 167.—Feathering Paddle-wheel.

occupy the least space, the rate of revolution should be as high as practicable. A high rate of slip is, therefore, altogether an advantage. In practice, therefore, it is quite usual to find the rate 20 per cent., and sometimes even as high as 25. For tug boats, however, 15 per cent. is general practice, as the float is not designed for the speed of the ship, but the power of the engine, although even in such ships the engines must not be so muzzled as to be unable to develop good power.

If D be the effective diameter of a paddle-wheel, which may be taken at the float centres of a feathering wheel, and at the centres of floats of a radial one.

A is the area of *one* float.

V ,, the velocity of float centres in feet per second $= 2\pi D \times \frac{R}{60}$.

R ,, the revolutions of the wheel per minute.

S ,, the speed of ship in knots per hour.

v ,, the velocity of ship in feet per second $= \frac{S \times 6{,}080}{60 \times 60} = 1{\cdot}689\,S$.

f ,, the fraction the slip is of V—that is, $\frac{V - v}{V}$.

e ,, the efficiency of the engines.

E ,, the efficiency combined of engines and wheels.

T_R ,, the resistance of the ship in pounds.

Then, Power transmitted to wheels = I.H.P. $\times\ e$.

Power delivered by the wheels = I.H.P. $\times$ E.

The stream of water projected by *each* wheel, $A \times V \times 64$ lbs.

The mass of water, $\frac{A \times V \times 64}{32} = 2A \times V$.

The acceleration given to the water is $(V - v)$.

Their pressure on the float $= 2A \times V\,(V - v)$.

(*a*) The net work done (thrust $\times$ speed of ship in feet per minute) $=$ $2A \times V\,(V - v) \times 60\,v = 120\,A \times V\,(V - v)\,v$.

But the work done is also measured by $T_R \times v$, and by I.H.P. $\times$ E.

That is, $\frac{T_R \times v}{33{,}000}$ = I.H.P. $\times$ E, or the tow rope horse-power (P_R H.P.).

The speed of the wheel, $V = \frac{\pi D \times R}{60}$; it is also equal the speed of the ship plus the slip—that is, $v + (V - v)$, from which may be deduced the rule that—

$$\text{Diameter of wheel in feet} = 19 \times \frac{V}{R}.$$

The Area of the Paddle Float can be calculated from first principles if it is true that the transverse section of the wheel race or stream of water projected from it is equal to the area of a float, and that the observed or apparent slip is the real slip. Assuming they are so, then—

Quantity of water projected per second $= \dfrac{A \times \pi D \times R}{60}$ cubic feet.

The acceleration may be taken as $fV = \dfrac{f \times \pi D R}{60}$.

Then mass of water $= \dfrac{A \times \pi D \times R}{60} \times \dfrac{64}{32} = \dfrac{A \times \pi D \times R}{30}$.

$$\text{Thrust} = \frac{A \times \pi D \times R}{30} \times \frac{f \times \pi D \times R}{60} = \frac{Af}{1{,}800}(\pi D \times R)^2.$$

or,

$$A = \frac{\text{thrust} \times 1{,}800}{f(\pi D \times R)^2} = \frac{\text{thrust}}{(D \times R)^2} \times \frac{182}{f}.$$

This thrust is that due to the effort of one wheel; so that if the thrust is calculated from the ship resistance, and there are two wheels, half the value so found must be inserted in the above equation.

Applying this theoretical formula to establish a rule to agree with the best practice of the day, the following holds good :—

$$\text{Area of one feathering float} = \frac{\text{I.H.P.}}{f - f^2} \times \left(\frac{C}{D \times R}\right)^3.$$

If E is 0·6, the value of C for *a pair* wheel ship in which I.H.P. is the *gross* power developed by the engine, 83·2.

If there is a *single* wheel at the stern, C = 109.

As some paddle-wheel steamers have an efficiency value as high as 0·66, the value of C is in that case 85·6; this, however, is not common, while, on the other hand, there are those whose efficiency is below 0·6.

Example (i.).—What should be the area of float of a side-wheel steamer having wheels 22 feet diameter revolving at 35 per minute, and a gross I.H.P. 4,150, the slip percentage being 15 and speed 20 knots?

$$\text{Here area of float} = \frac{4{,}150}{0{\cdot}15 - 0{\cdot}15^2} \times \left(\frac{83{\cdot}2}{22 \times 35}\right)^2 = 41{\cdot}5 \text{ square feet.}$$

Example (ii.).—What float area should a stern wheeler have with a wheel 10 feet diameter running at 50 revolutions with a speed of 12 knots, the slip 20 per cent., and I.H.P. 400.?

$$\text{Area float} = \frac{400}{0{\cdot}2 - (0{\cdot}2)^2} \times \left(\frac{109}{10 \times 50}\right)^3 = 25{\cdot}8 \text{ square feet.}$$

The Number of Floats on a Feathering-wheel should be such that when revolving at slow speeds there is not too much thud owing to irregular angular advance—that is to say, the floats must be near enough together to maintain something like an uniform area of float surface acting on the water.

Rule in Practice.—Number of floats $= \dfrac{D + 2}{2}$.

Radial wheels usually had a float for each foot of diameter, which means they were at the tips 3·14 feet apart. The tendency of modern practice

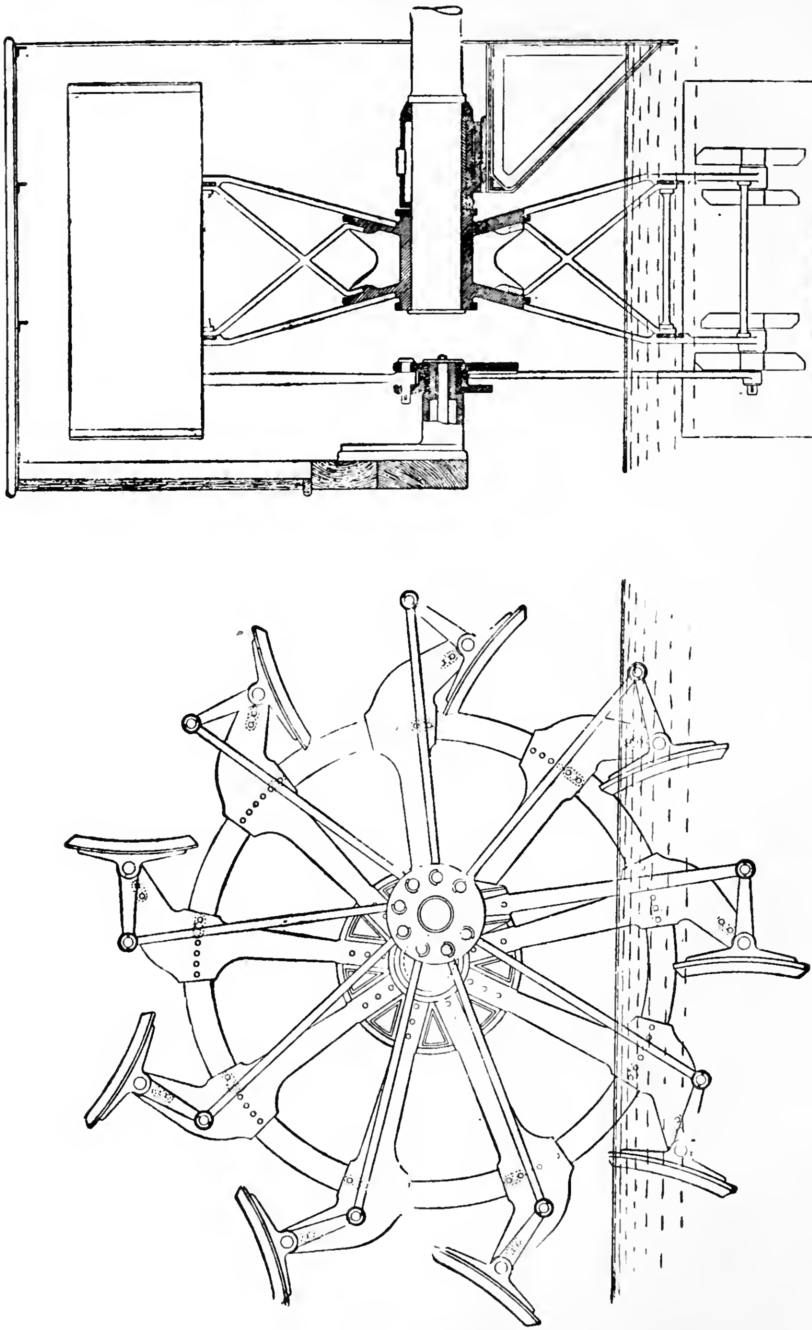


Fig. 168.—Feathering Paddle-wheel with Steel Floats.

with the higher rate of revolution is to have as few floats as consistent with good working.

The Proportions of Paddle Floats are quite arbitrary within reasonable bounds. In practice the ratio of length to breadth on a radial fixed float is 4 to 5, while for a feathering-wheel it is 2·6 to 3—that is, $A = r B \times B$, or $r B^2$, where B is the breadth in feet and r is the ratio; hence

$$\text{Breadth of float} = \sqrt{\frac{A}{r}}.$$

Since length is r B, then

$$\text{Length of float} = r\sqrt{\frac{A}{r}}.$$

For example, take the float at 41·5 square feet, and r as 3—

Then, $$\text{Breadth of float} = \sqrt{\frac{41 \cdot 5}{3}} = 3 \cdot 72 \text{ feet.}$$

$$\text{Length} = 3 \times 3 \cdot 72 = 11 \cdot 16 \text{ feet.}$$

If the steamer has to navigate narrow passages, as dock entrances, etc., the ratio will not exceed 2·6, in that case—

$$\text{Breadth of float} = \sqrt{\frac{41 \cdot 5}{2 \cdot 6}} = 4 \cdot 0 \text{ feet.}$$

$$\text{Length} \quad ,, \quad = 4 \times 2 \cdot 6 = 10 \cdot 4 \text{ feet.}$$

The Thickness of the Elm Floats is usually about $\frac{1}{12}$ the breadth, but steel floats, as shown in fig. 168, made of plate stiffened by slightly curving it. The edge resistance of steel plates is, of course, less than that of elm boards, but, if damaged by contact with wreckage, they are not so easy to remove in a seaway, although when removed they can be generally restored to proper shape and used again. The steel plates are also cheaper, as there is considerable labour involved in preparing, shaping, and bolting together the elm when of large size.

Paddle-Wheels are usually made now without outer rims, as shown in fig. 168; they are thereby less liable to damage, and the limit of extreme diameter is less; moreover, the friction in the water is less, although to a large extent any resistance to the passage of the wheel should go towards thrust. On the other hand, the floats have not the same protection as they enjoy with the outer rim, and the arms beyond the inner rim being cantilevers, must be much stouter than when the arm has the double support of the two rims. The structure of wheel is now wholly of steel, as being cheaper and, on the whole, stronger than wrought iron; and the hub or boss, which was formerly of cast iron hooped with wrought iron, is now often a steel casting.

The Feathering Centre and Gear is, for convenience, often secured to the sponson beam, as shown on fig. 168. but, as this beam is liable to be sprung considerably when coming alongside piers, especially in rough weather, it is safer and snugger to fit an eccentric on the outer bearing, as in fig. 169. This arrangement possesses another advantage that, being accessible and somewhat shielded from water wash, it can be attended to and treated with

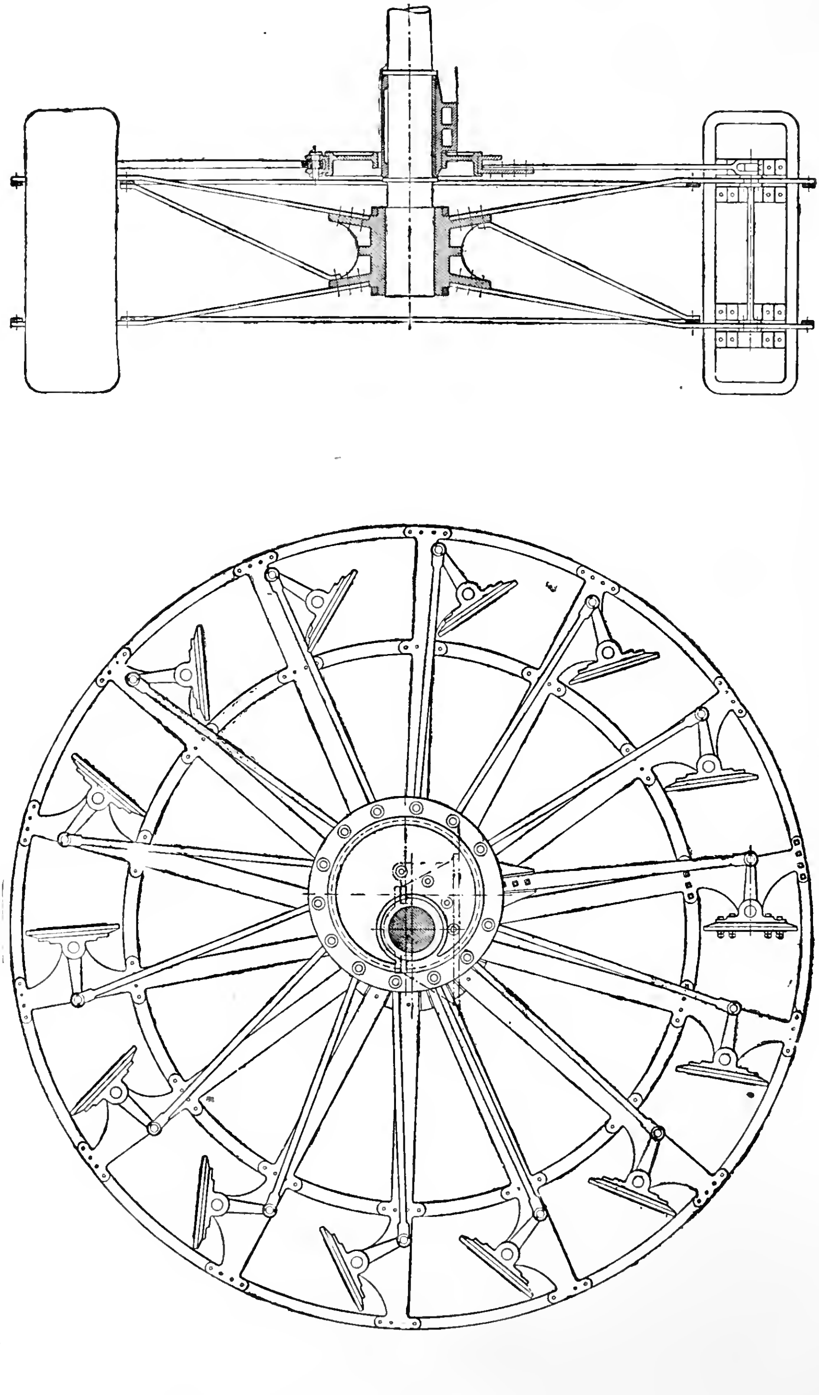


Fig. 169.—Feathering Paddle-wheel, Feathering Gear inside.

oil as a lubricant. The eccentric is rigid in its connection, and has no spring in itself, and liable to no damage as the outer gear is.

The Immersion of the Paddle Float must be greater in sea-going ships than in those limited to smooth water. In a general way ships which are liable to ply among waves should have in smooth water an immersion of the inner edge of float when at lowest position equal to one-half the breadth, while if for service in comparatively smooth water an immersion of a quarter is sufficient. For light-draft steamers as little as one-eighth suffices, but it must always be borne in mind that the greater the immersion the larger is the section of wheel race: moreover, as the immersion is increased by the additions to the deadweight carried, some regard must be given to the circumstances of the ship's lading.

Paddle-wheel Bosses are made of the form shown in figs. 167, 168, and 169, and formerly of the toughest cast iron; they were, when of large size, further strengthened against splitting by shrinking on wrought-iron hoops, as shown in fig. 168, both at the centred body and at the edges of the flanges. To-day such castings can be obtained in steel which require no such safeguards, and are much lighter as well as stronger. The earliest practice was to stake the boss on the square section end of the paddle-shaft; that, however, is a bygone practice, which, while rough and ready, had the merit of safety. Since then the boss has been bored out to fit the shaft, and keyed to it with two keys 90° apart. It was, however, most difficult to withdraw bosses fitted in this way; therefore, the better plan is to fit them with a taper, as is done with screw propeller bosses.

Taking the diameter of the shaft, which is sufficient for the torque of the engines, as a criterion of size from which to calculate the wheel scantlings, or take the diameter of the inner journal of the actual shaft calculated by the following, viz. :—

$$\text{Diameter of inner journal} = \sqrt{\frac{\text{I.H.P.}}{\text{R}} \times \text{F}}.$$

For a side-wheel steamer with a single cylinder, .			. .	F = 80.
For side wheels with two-cylinders, cranks at 90°,			. .	F = 58.
,,	,, ,, ,,	and an intermediate shaft,		F = 50.
,,	,, ,, ,,	2 solid cranks, 90°, .	.	F = 55.
,,	,, three- ,,	3 cranks, 120°,	. .	F = 50.

Thickness of cast-iron boss round shaft		$= 0{\cdot}28 \times d.$
,, steel ,, ,,		$= 0{\cdot}20 \times d.$
Diameter of outer flanges,		$= 4{\cdot}0 \times d.$
Thickness ,,	cast iron	$= 0{\cdot}15 \times d.$
,, ,,	cast steel	$= 0{\cdot}11 \times d.$
Breadth of two hard steel keys each		$= 0{\cdot}18 \times d + \frac{1}{4}$ inch.
Thickness ,, ,, ,,		$= 0{\cdot}09 \times d + \frac{1}{4}$ inch.

All shafts over 8 inches diameter should have two keys.

Paddle Arms of small radial wheels with two rims are generally made of flat rolled bars. In large feathering wheels they are always forgings and now generally of steel.

Let n be the number of floats on a wheel, with a pair of arms to each float, then if the framework is braced and tied together as a compact structure, and thereby distribute the load, and b is the breadth and t the thickness of each arm at the boss—

$$\text{The resistance to bending of each arm is} = f(t \times b^2).$$
$$\text{The total resistance of the structure} = 2nf\,(t \times b^2).$$

The shaft journal has to work against this resistance, which may be designated as P.

Then $\dfrac{\text{P} \times \text{diameter of wheel}}{2}$ is the torsion on the shaft, and equal to the bending moment on the wheel structure.

Now, $\quad$ the torque $= \dfrac{\pi d^3}{16} \times f_1$.

Hence $\quad \text{P} \times \dfrac{\text{D}}{2} = \dfrac{\pi d^3}{16} \times f_1 = 2nf_1\,(t \times b^2).$

If f_1 be taken as practically equal to $0{\cdot}8f$ for ordinary mild steel, then

$$t \times b^2 = {\cdot}0982\,d^3 \times 0{\cdot}8 = 0{\cdot}07854\,\frac{d^3}{n}.$$

In practice the ratio of b to t is 5 at the boss and 3·5 with same thickness at the rim; when there is only an inner ring the ratio of b to t just outside the ring is from 6 to 7.

Taking the ratio at boss as 5, then $t \times b^2 = \dfrac{b^3}{5}$.

$$\text{Then } \frac{b^3}{5} = 0{\cdot}07854\,\frac{d^3}{n}\,;\ \text{and } b = d\sqrt[3]{\frac{{\cdot}3928}{n}}.$$

$$\text{Then breadth of arm} = \frac{0{\cdot}73\,d}{\sqrt[3]{n}}.$$

For example, if n is 8, then $b = 0{\cdot}365d$.

For tug boats and ships making sea passages in winter time, where the wheels are liable to heavy shocks from the sea, the structure should be a somewhat heavier one; moreover, in ordinary work the wheel is liable to rough usage, and the load is not evenly distributed. To provide for such contingencies, the divisor should be $(n - 2)$ rather than n.

The Rims of Paddle-wheels are of rolled bar iron or mild steel, and when there are two rims, as in fig. 167—

$$\begin{aligned}
\text{Breadth of outer ring} &= 0{\cdot}4 \times d.\\
\text{,, \quad inner ,,} &= 0{\cdot}4 \times d.\\
\text{Thickness of outer ,,} &= 0{\cdot}08 \times d.\\
\text{,, \quad inner ,,} &= 0{\cdot}10 \times d.
\end{aligned}$$

When there is only an inner ring the section should be $0{\cdot}5d \times 0{\cdot}14d$ with bolts $0{\cdot}15d$ in diameter, while the bolts with two rings may be smaller, or $0{\cdot}12d$. The diagonal tie-rods are $(0{\cdot}18d + \frac{1}{2}$ inch) in diameter, while the cross bars are $(0{\cdot}18d + \frac{3}{4}$ inch), and made hollow.

It should be noted that these rules are laid down for guidance, for in actual practice the nearest standard section should be selected for use, as also the bolts used should be of store dimensions.

Of the Feathering Gear, the float gudgeons should be in diameter equal to (0·1 the breadth of float + $\frac{1}{2}$ inch), and the length of bush 1·40 the diameter.

The radius-rod pins should be in diameter = 0·50 that of the gudgeons, and fitted as shown in fig. 169. The pins, gudgeons, etc., are usually cased in hard bronze, and the holes in which they fit bushed with *lignum vitæ*, inasmuch as the only lubricant they get is water. It is, however, not uncommon now to use hard steel or case-hardened pins, etc., and employ white metal bushes, which wears well in sandy water.

The Position of the Gudgeons on the Floats is not always the middle, but nearer to the centre of pressure, as shown in fig. 167. About two-fifths the breadth of the float from the outer edge is a good position for the turning axis, as then the twisting load on entry is not so great, and during the progress through the water there is a better balance than if hung centrally.

The Outer Bearing of the Paddle-shaft has to take the weight of the shaft and wheel and the horizontal thrust of the propeller; it ought, therefore, to be designed with the centre line diagonal instead of vertical, as it usually is for convenience of getting the shaft into place. The resultant pressure on the bearing is diagonal in direction and of considerable magnitude, so that the bearing must be large and strong. It is usually two diameters of the shaft-journal in length, and lined with white metal. It is largely treated with water, so that it cannot get hot, and supplied with a central chamber filled with tallow, which keeps out the water and lubricates the bearing if it gets warm. It is desirable that it shall be also lubricated with oil, as is any other bearing, if good results are to be expected. The cap of the bearing is generally a plain iron casting like that on a tunnel bearing in a screw ship. It takes no load, and only prevents the shaft from lifting or jumping in the bearing, as it might do in a seaway.

The Screw Propeller is a more important instrument of propulsion, as it is in universal use for every kind of ship and on every service, having ousted the paddle in places where at one time it seemed impossible to be employed with advantage. The inventions of Sir John Thornycroft and Sir A. Yarrow, whereby screws can be employed in the shallowest of water—so shallow, indeed, that when at rest the water level is very near the screw axis—have revolutionised the practice of marine engineers. Moreover, the covering in of the propeller races in the ways adopted by these gentlemen has immensely improved the efficiency of all screw ships where the screw was quite but only just immersed.

For Purposes of Calculation it is sufficient to assume that the stream of water from a screw is a hollow column of the same external diameter as the screw, and the internal is as the diameter of the boss. In the mercantile marine the boss is so small, compared with the diameter of the screw, especially with solid ones, that no great discrepancy arises if the column is assumed to be solid. In the naval service, and with all ships having large power and screws of small diameter, this course cannot be adopted, for the boss disc is from 5 to 12 per cent. of the screw disc.

The Surface itself is now generally a portion of a true helix, although there are still not wanting those who hold that there is some virtue in a screw of variable pitch, and while certain incline to a variation from the boss to the tip, others prefer the increase to be from the leading edge to the following, so that the water at contact with the blade may have only a very slight displacement, but a gradual increase as it passes along the face with the final acceleration just as it leaves at the following edge. If such propellers continue to be made, the effective pitch, or that taken for calculation, must not be a mean, but the final pitch.

The Original Screw of F. P. Smith (*v.* fig. 170), as adopted in the Navy and mercantile marine, was a true screw of two convolutions cut off by parallel planes one-third to one-eighth of the pitch apart; in fact, it was a two-bladed screw with very wide tips, but quite a moderate amount of acting surface, so that in H.M.S. "Rattler" it was very efficient, much more so than most of the screws made for many years afterwards and called "common screws," but of very large diameter, and their length only one-eighth the pitch.

Theoretically, the Two-bladed Screw is the most efficient, and in practice in smooth water on a ship with fine lines it is likewise so; the vibration set up by it, however, militates so much against its use that to-day it is only found in quite small craft, and in those ships that still have to make use of sail power as well as steam.

The Developed Surface of a screw is that actually acting on the water

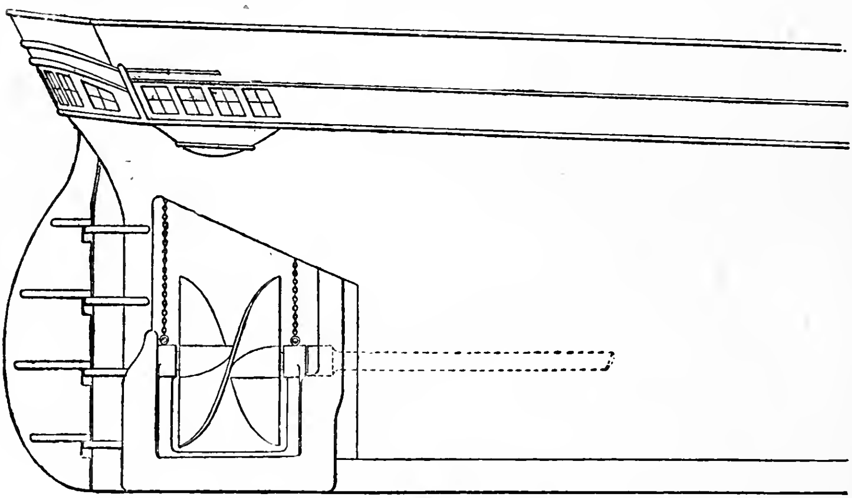


Fig. 170.—Smith's Screw Propeller as first fitted in the Navy.

and capable of being measured as a part of the blade. It is, therefore, sometimes called the acting surface.

The Projected Surface is that of the blade as projected on a plane at right angles to the axis, or the view of the blade from abaft as drawn on paper. This is sometimes spoken of as the *active surface*, and taken as the criterion of the screw's ability to produce thrust, being the transverse component of each portion of the real surface.

The Diameter of the Screw is that of the circle moved through by the extreme tip of a blade.

The behaviour of a screw and the conditions prevailing in its neighbourhood when at work have been already dealt with (Chap. ii.), it will, therefore, be assumed here that the screw is working in practically undisturbed water, which before entering the screw race is moving with respect to the screw at the same speed as the ship, but in the opposite direction. This is really the condition under which model screws generally have been tried in the past. To-day, with the modern tank and its elaborate apparatus, the model screw is tried behind the model ship, which is presumed to have similar wake currents to that of the real ship.

The Thrust of a Screw is the resultant of all the pressures on it acting in a direction parallel to the axis applied axially through the shafting to the thrust block.

Indicated Thrust is an approximation to the actual thrust arrived at by means of a formula proposed by the late Dr. W. Froude, and very useful for guidance in dealing with propeller problems when rightly understood and properly applied. It can be used with advantage in treating comparatively the results of trials, but for absolute results it cannot be relied on. Froude's rule is as follows:—

$$\text{Indicated thrust in pounds} = \frac{\text{I.H.P.} \times 33{,}000}{\text{V} \times 60},$$

where V is the velocity of the stream in feet per second — that is = pitch × revolutions.

With the 80 per cent. efficiency of the engines in Dr. Froude's day the results obtained by this formula were more nearly true than now with better engines.

$$\text{Real thrust} \times 60\, v = \text{I.H.P.} \times \mathbf{E} \times 33{,}000.$$

If $\frac{v}{\text{V}} = \text{E}$, then Froude's formula holds good—that is, when the slip per cent. is the same as the per cent. of loss of engines and screw it is true.

The efficiency of the screw may be examined on the system proposed by him—viz., by constructing curves of *indicated thrust* on the same principle as that described in Chap. ii., for curves of I.H.P.

In this case, however, the ordinates represent the thrust as calculated from the I.H.P., or from the pressure on the pistons, and this for convenience is generally expressed in tons.

It is assumed that the pressure on the pistons multiplied by twice their stroke is equal to the thrust multiplied by the pitch, and if there were no loss by friction of machinery, etc., by the principle of work this would be true.

Let p be the mean effective referred pressure on the low-pressure pistons in pounds per square inch, n their number, A their area in square inches, L the length of stroke in feet, and P the pitch of the screw in feet; then,

$$\text{Thrust} \times \text{P} = p \times \text{A} \times n \times 2\,\text{L},$$

or

$$\text{Thrust} = \frac{p \times \text{A} \times n \times 2\,\text{L}}{\text{P}}.$$

If both numerator and denominator of the fraction be multiplied by R, the number of revolutions per minute,

$$\text{Thrust} = \frac{p \times \text{A} \times n \times 2\,\text{L} \times \text{R}}{\text{P} \times \text{R}};$$

but

$$\frac{p \times \text{A} \times n \times 2\,\text{L} \times \text{R}}{33{,}000} = \text{I.H.P.}$$

Therefore

$$\text{Thrust} = \frac{\text{I.H.P.} \times 33{,}000}{\text{P} \times \text{R}}.$$

This is called the *indicated thrust*, and it was by constructing a curve of *indicated thrust* that the inefficiency of the original screws of H.M.S. "Iris" was discovered.

Dr. Froude* also explained the method by which he estimated the "*initial friction*," or "the equivalent of friction of the engines due to the working load." He said—"When decomposed into its constituent parts indicated thrust is resolved into several elements, which must be enumerated and kept in view. These elements are—1, the useful thrust, or ship's true resistance; 2, the augment of resistance, which is due to the diminution which the action of the propeller creates in the pressure of the water against the stern end of the ship; 3, the equivalent of the friction of the screw blades in their edgeway motion through the water; 4, the equivalent of the friction due to the dead weight of the working parts, piston packings, and the like, which constitute the initial or slow-speed friction of the engine; 5, the equivalent of friction of the engines due to the working load; 6, the equivalent of air-pump and feed-pump duty.

"It is probable that 2, 3, and 4 of the above list are all very nearly proportional to the useful thrust; 6 is probably nearly proportional to the square of the number of revolutions, and thus, at least at the lower speeds, approximately to the useful thrust; 5 probably remains constant at all speeds, and for convenience it may be regarded as constant, though perhaps in strict truth it should be termed 'initial friction.' If, then, we could separate the quasi-constant friction from the indicated thrust throughout, the remainder would be approximately proportional to the ship's true resistance.

"Now, on drawing a curve (of indicated thrust) . . . it becomes at once manifest in every case, that at its low-speed end the curve refuses to descend to the thrust zero, but tends towards a point representing a considerable amount of thrust, and it is impossible to doubt that this apparent thrust at the zero of speed, when there can be no real thrust, is the equivalent of what I have termed initial friction; so that if we could determine correctly the point at which the curve, if prolonged to the speed zero, would intersect the axis O Y (fig. 171), and if we were to draw a line through the intersection parallel to the base, the height which would be thus cut off from the thrust ordinates would represent the deduction to be made from them in respect of constant or initial friction, and the remainders of the ordinates between this new base and the curve would be approximately proportional to the ship's true resistance."

Dr. Froude then explains his method, which is substantially as follows:—

Let O B, O C, O D represent the three progressive speeds, observed in the usual way, and B E, C F, D G the indicated thrust, calculated from the data observed at those speeds; through the points G, F, E draw the curve of indicated thrust. Let O A represent a low speed, which should not exceed 5 knots, and A H the corresponding indicated thrust; continue the curve to H, and at H draw a tangent to the curve cutting O Y. Take a point M between O A, so that $\frac{\text{O M}}{\text{M A}} = \frac{0{\cdot}87}{1{\cdot}0}$. Draw M K parallel to A H, and cutting the tangent line at K. Through K draw a line K T parallel to O D, cutting

* *Transactions of the Institution of Naval Architects*, vol. xvii., 1876

O Y at T. Then O T is the part cut off O Y by the curve, and represents the initial friction.

Dr. Froude deduced, from careful investigation, that only 37 to 40 per cent. of the whole power delivered is usefully employed, and that "the constant friction is equivalent to from one-eighth to one-sixth of the gross load on the engine when working at its maximum speed and power."

Rankine's Rule for Thrust is summarised as follows :—

Professor Rankine gives (*Rules and Tables*, p. 275) this same rule in another and more convenient form for practical use, viz. :—

RULE V.—*To calculate the thrust of a propelling instrument (jet, paddle, or screw) in pounds; multiply together the transverse sectional area, in square feet, of the stream driven astern by the propeller; the speed of the stream relatively to the ship in knots; the real slip, or part of that speed which is impressed on*

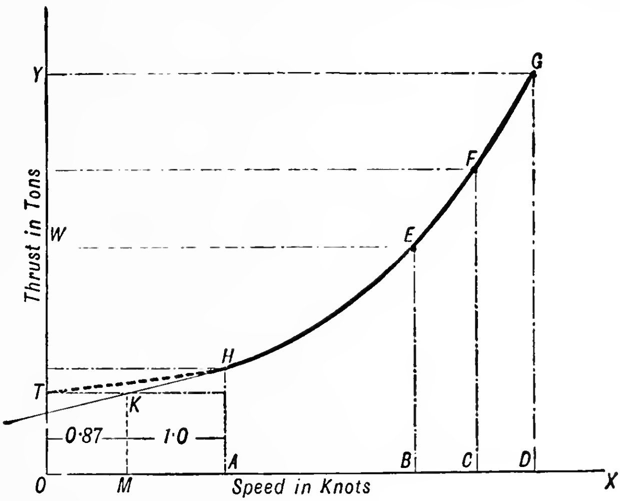


Fig. 171.—Curve of Indicated Thrust.

that stream by the propeller, also in knots; and the constant 5·66 *for sea-water, or* 5·5 *for fresh water.*

That is, if S is the speed of the screw in knots, s the speed of the ship in knots, A the area of the stream in square feet (of sea-water),

$$\text{Thrust in pounds} = A \times S(S - s) \times 5{\cdot}66, \text{ or } D^2 \times S(S - s) \times 4{\cdot}45.$$

The effective slip or the actual mean velocity imparted to the water cannot be found accurately any more than can the mass of water set in motion. The effective thrust, however, may be found approximately, thus—

Rule.— Effective thrust $= (D \times S)^2 \div F$.

D the diameter in feet, S the speed of screw in feet per second, and F varies from 3·3 with small quick propellers to 5 with large slow ones. Ordinary merchant steamers, 5 to 4·5; fast-running merchant and naval ships. 4·0; very high speed, as with turbines, 3·3.

TABLE XLVII.—Particulars of Screw Propellers.

Ship. S = Single Screw. T = Twin Screw.	Diam.	Pitch.	Blades.				Length of Boss.	Diam. of Shaft.	I.H.P./R	Material.		Weight.
			No.	Max. Breadth.	Thick.	Total Surface.				Boss.	Blades.	
Solid Cast Iron.												
	Ft. Ins.	Ft. Ins.		Ft. Ins.	Ins.	Sq. ft.	Ft. Ins.	Ins.				Cwts.
Dn., S.,	19 6	26 6	4	3 10½	10	112	3 9	17¾	64·7	...	...	250·5
Nno., S.,	18 3	26 0	4	3 9	9	96	3 0	13	36·1	...	...	202·5
Slr., S.,	16 8	23 6	4	4 3	8½	92	3 6	13¾	36·7	...	...	189·0
Ddo., S.,	17 6	17 6	4	3 6	7½	90	2 9	13½	34·2	...	...	124·7
Ard., S.,	16 0	20 0	4	3 9½	7½	85	2 9	13¾	33·0	...	...	121·0
Arl., S.,	16 0	18 0	4	2 9¼	7¼	74	2 9	12¼	21·8	...	...	110·75
Eld., S.,	13 6	21 0	4	3 3¾	7¼	68	2 9	11½	28·8	...	...	98·90
Hyd., S.,	12 9	16 0	4	3 3½	6½	58	2 5	11	14·8	...	...	75·65
Esw., S.,	13 0	16 4	4	2 5½	5¾	48	2 3	9¾	10·3	...	...	61·40
Alt., S.,	10 3	17 3	4	2 11¼	5	43	2 1	8⅞	8·7	...	...	44·10
Bln., T.,	12 0	16 6	3	3 3	8	40·5	2 6	11¾	20·0	...	...	75·50
Qnw., S.,	9 9	13 6	4	2 4	4	33·5	1 9½	6⅞	6·10	...	...	29·60
Hwh., T.,	9 0	14 0	3	2 10¾	4¾	26·0	1 10	7¾	5·69	...	...	27·50
Erp., S.,	9 0	13 0	3	2 2½	4½	21·0	1 6	6⅞	4·00	...	...	20·30
Wtm., S.,	7 8	9 0	3	2 1½	3½	17·5	1 3	5½	2·30	...	...	12·80
Skb., S.,	6 3	7 0	3	1 9¼	3¼	12·5	1 1½	4¾	1·15	...	...	8·50
Fbc., S.,	4 6	5 0	3	1 1¾	2½	5·5	0 9	3⅝	0·42	...	...	2·90
Loose Blades Bolted On.												
	Ft. Ins.	Ft. Ins.		Ft. Ins.	Ins.	Sq. ft.	Ft. Ins.	Ins.				
Cpt., S.,	20 3	25 0	4	4 5¼	8½	120	3 9	18¼	70·0	C. steel	M. bronze	311·5
Mtl., S.,	19 0	22 0	4	3 8½	6	96·5	3 6	15½	40·0	C. steel	Steel	196·7
Str., S.,	19 0	23 0	4	4 4½	8¼	96·0	3 7½	15½	41·2	C. iron	C. iron	288·0
Czg., S.,	17 0	21 6	4	4 2	9¼	90·0	3 6	15	41·0	C. iron	C. iron	223·9
Ldn., T.,	17 6	19 9	4	4 0	7¼	86·0	4 0	17½	72·0	M. bronze	M. bronze	261·0
Std., T.,	16 0	25 0	3	3 1¾	6½	77·0	3 9	18	75·0	D. metal	D. metal	216·5
Stm., S.,	17 0	21 0	4	3 5¾	7⅜	72·0	2 8	13½	30·1	C. iron	C. iron	183·7
Rmo., S.,	15 0	23 0	4	3 8¾	6¾	63·6	2 9	13½	25·0	C. iron	C. iron	170·3
End., T.,	16 6	22 0	3	3 10	7¼	60·0	3 4½	16⅛	54·5	A. bronze	A. bronze	166·6
Kvn., S.,	14 0	17 0	4	2 9	6	55·0	2 1	9¾	11 2	C. iron	C. iron	99 5
Apl., T.,	13 0	17 0	3	3 3½	6¼	40·0	3 0	14½	32·9	A. bronze	A. bronze	107·7
Clm., T.,	12 0	16 0	3	3 2¼	5	39·0	2 6	11¾	20·1	C. steel	S. bronze	71 80
Cbg., T.,	11 6	18 6	3	3 1¼	4¼	34·0	2 2	10¼	13·3	C. iron	S. bronze	62·85
Prl., T.,	12 6	14 0	3	2 9	6	33·0	2 6	12⅛	22·8	A. bronze	A. bronze	76·5
Prs., T.,	10 6	12 0	3	2 5¼	4⅞	24·0	2 6	12⅝	16·3	A. bronze	S. bronze	64·7
Blc., T.,	9 3	10 0	3	2 5¼	3¾	21·3	2 1½	9	7·5	A. bronze	A. bronze	36·0
Slm., T.,	6 7	8 9	3	1 10¾	2	12·7	1 0¾	6½	5·0	S. bronze	S. bronze	7·45

The Pressure per Square Inch on a Propeller Blade at any position and at any rate of revolution may be found by referring to the diagram on p. 479 (fig. 172). If, for example, it is required to know the pressure at a radius of 6 feet when the revolutions are 80, all that is necessary is to make the intersection of the vertical line through 6 with the curve for 80 revolutions,

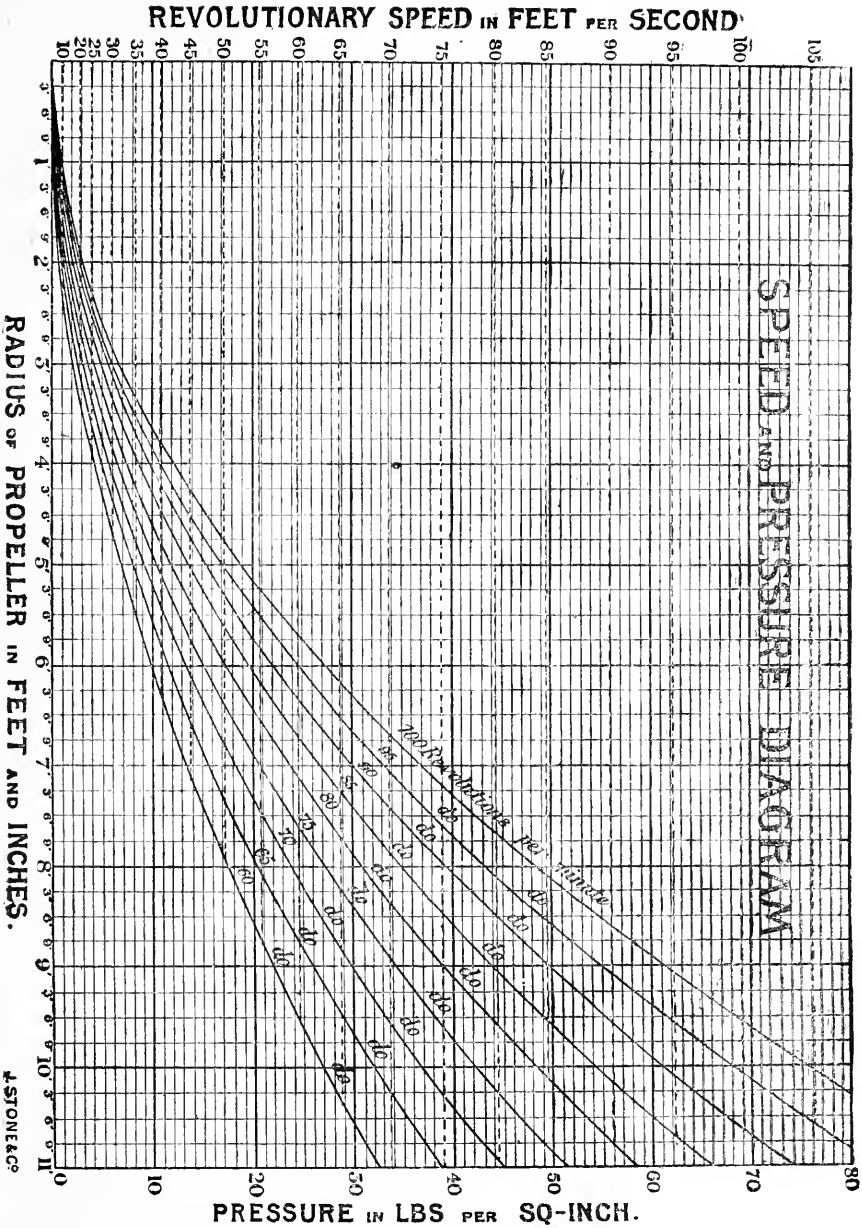


Fig. 172.

and the horizontal line through it to the right-hand side shows it to be a little over 17 lbs.

The Diameter of Screw best suited to a Ship is a matter for much consideration, and to arrive at it all the circumstances of the particular case must be taken into account. The three leading features of a screw—viz., diameter, pitch, and surface of blades—are interdependent, one on each of the other two, or two together on one. The shape of the ship and her draught of water are governing factors; and, last but not least, there must be some relation between the power developed per revolution by the engine or motor and the area of disc of the propeller, as also the area of blades themselves. There are some other considerations extraneous to ship and engine, such as the fear of cavitation with a coarse pitch and insufficient blade surface; the tendency to swamp the screw with air if the blade tips are too near the surface; and the risk of damage if the blades of multiple-screw ships project so far out sideways that the quarters of the ship cannot protect them from contact with dock and quay walls.

In a general way it may be taken as an axiom that the screw of highest efficiency is the one which with the smallest diameter is sufficient for the power transmitted to it. The objection to large diameter screws for high efficiency of propulsion as against high thrust is well set forth by Dr. W. Froude, who says—

"Take the case of a screw 20 feet diameter, making 80 revolutions per minute; the tips of the blades are travelling at a speed of about 50 knots; now, the resistance of a surface so short in the line of motion as a screw blade, *even when its surface is quite smooth*, is as much as $1\frac{1}{4}$ lbs. per foot at 10 knots, and is nearly as the square of the speed, and as each square foot of blade area involves 2 square feet of skin, the resistance of each is over 60 lbs.; thus, making some allowance for thickness and bluntness, there is involved in driving it at 50 knots at least 10 I.H.P., and collectively the outmost foot of four such blades, each 3 feet wide, would absorb fully 120 I.H.P. in surface friction; and though the parts nearer to the root move with proportionally less speed, and therefore with less resistance, yet, on the other hand, screw blades are generally rough from the sand, and have probably a still higher coefficient of frictional resistance."

If a screw is of sufficient diameter, and has ample blade area, any addition beyond the tip will act then quite as a brake and seriously reduce its efficiency; and that such screws produce augmented resistance is also a common experience.

The loss by surface friction of blades in these days of high rates of revolution is more pronounced than ever. But perhaps the most attractive reason for adopting the small screw is that greater immersion of upper-blade tip is possible; all other things being equal, the small screw thoroughly well immersed must be a more efficient one, in a sea-going ship especially, than a larger one with small immersion.

That the Limit to Screw Diameter is the draught of water of the ship at the stern no longer holds good as it did when draft of water was the governing factor. As already shown, a screw in a shallow-draft ship may be largely in excess of it if means are taken to keep the water massed up by the screw when working from the pressure of the atmosphere and from dispersal. Professor Flamm has shown how, by means of a stream of water at high velocity, directed diagonally along a surface, the relief of air pressure is greater than the normal effect of the impinging stream on that surface.

It is also a matter of common knowledge that, if the stern lines of a ship are full, so that the flow of water into the screw race is restricted, a small diameter screw is a very inefficient one to produce thrust, so the ordinary cargo steamer must have one of large diameter compared with the engine size to work effectually. On the other hand, a twin-screw ship having its propellers in the open, and largely free from the influence of ordinary wake currents, may have screws of smaller diameter in proportion to the engine power.

It is evident then the diameter must be such as to suit both the ship and the engine power, and that it cannot be arrived at satisfactorily by considering only the needs of the one, as was the case when the ratio of area of screw disc to that of immersed mid-section was the criterion. The following rules may be used to give an appropriate diameter of screw under the varying circumstances experienced in ship design :—

D is the diameter of the L.P. cylinder of the engine whose piston stroke is S, both in feet.

P_c is the prismatic coefficient of the ship $= \dfrac{\text{displacement} \times 35}{\text{length} \times \text{mid-section}}$.

Z a multiplier $= (2{\cdot}4 + P_c)$ for twin screws,
and $(2{\cdot}8 + P_c)$ for single screws.

R the revolutions per minute of the screw.

RULE I.—Diameter of screw in feet $= Z \times \sqrt{D \times S}$.

RULE II.— „ „ $= x \times P_c \sqrt[3]{\dfrac{\text{I.H.P.}}{R}}$.

For single screws, ...	$x = 7{\cdot}25$.	Ocean-going express, 7·61.
„ twin screws, ...	$x = 6{\cdot}55$.	„ „ 6·88.
„ quadruple screws, . . .	$x = 6{\cdot}25$.	„ „ 6·51.
* „ turbine-driven centre screw,	$x = 6{\cdot}55$.	„ „ 6·88.
„ „ wing „	$x = 5{\cdot}75$.	„ „ 6·04.

N.B.—In no case, however, must P_c have a less value than 0·55.

Example (i.).—What should be the diameter of the screw of a cargo boat having engines with cylinders 25, 38, and 64 inches diameter and 45 inches length of stroke, the prismatic coefficient of hull being 0·85 ?

$$\text{Diameter screw (Rule I.)} = (2{\cdot}8 + 0{\cdot}85) \times \sqrt{\frac{64}{12} \times \frac{45}{12}} = 16{\cdot}3 \text{ feet}$$

Example (ii.).—An express steamer for cross-channel service is intended to be twin screw and to have engines of 6,000 I.H.P. aggregate. The rate of revolution is 180, and the prismatic coefficient of hull is 0·6.

$$\text{Diameter of screw (Rule II.)} = 6{\cdot}55 \times 0{\cdot}6 \sqrt[3]{\frac{3{,}000}{180}} = 10{\cdot}02 \text{ feet.}$$

Example (iii.).—An ocean-going steamer driven by turbines is to have three screws, the total S.H.P. is 24,000 at 250 revolutions, the hull prismatic

* With geared turbines and electrically driven screws the ordinary rules hold good.

coefficient is 0·65, what diameter should be the screws, the power being evenly divided?

$$\text{Here, centre screw diameter} = 6{\cdot}88 \times 0{\cdot}65 \sqrt[3]{\frac{8{,}000}{250}} = 14{\cdot}17 \text{ feet.}$$

$$\text{wing} \quad ,, \quad ,, \quad = 6{\cdot}04 \times 0{\cdot}65 \sqrt[3]{\frac{8{,}000}{250}} = 12{\cdot}44 \text{ feet.}$$

The Diameter of a Screw Propeller may be determined, however, by means of a formula based on quite correct principles of theory, but with a factor or constant multiplier modified by applying the rule in its purely theoretical form to the actual successful practice of to-day with all kinds of ships.

(*a*) The thrust of a screw in pounds $= 2\,A \times V\,(V - v)$.

(*b*) The work done per minute $= 2\,A \times V\,(V - v)\,60\,v$ foot-lbs.

$$\text{The horse-power} = \frac{2\,A \times V\,(V - v)\,60\,v}{33{,}000} = \frac{A \times V\,(V - v)\,v}{275}.$$

If, as before, E is the efficiency of screw and engine—

$$(c) \qquad \text{I.H.P.} = \frac{A \times V\,(V - v)\,v}{275\,E}.$$

$$\text{Assuming} \qquad A = \frac{\pi\,D^2}{4}, \text{ or } 0{\cdot}7854\,D^2;$$

and $(V - v) \div V$ as the fraction of V for the slip, say f, so that—

$$\frac{V - v}{V} = f; \text{ or } (V - v) = fV,$$

$$\text{and} \qquad v = V - fV, \text{ or } V\,(1 - f).$$

Then, substituting these values in equation (*c*)—

$$\text{I.H.P.} = 0{\cdot}7854\,\frac{D^2 \times V \times fV \times V}{275\,E}\,(1 - f)$$

$$= \frac{D^2 \times V^3\,(1 - f)\,f}{350\,E}.$$

$$\text{But} \qquad V = P \times R; \text{ then I.H.P.} = \frac{D^2 \times (P \times R)^3\,(f - f^2)}{350\,E}.$$

In actual practice there are disturbing causes which prevent the factor from being so low as 350; for example, with very large bosses the column of water is a hollow one, and the equivalent diameter is then less than D. Moreover, the apparent slip f is less than the real or actual slip—that is, less than the true measure of the acceleration, for there are wake and other currents; and consequently to know how large a real screw should be for the needs in practice another factor is necessary; hence

The Rule for the Diameter of Screw for good work and high efficiency can be found by making the formula as follows:—

$$\text{Diameter of screw} = \sqrt{\frac{\text{I.H.P.}}{x\,(1 - x)} \times \left(\frac{C}{P \times R}\right)^3}, \text{ or } \sqrt{\frac{\text{I.H.P.}}{x - x^2} \times \left(\frac{C}{P \times R}\right)^3}.$$

$x = 0{\cdot}2\,P_c + f$, or an approximation to the real slip.

For single-screw ships and the centre screw of triple screws, $x = 0{\cdot}18\,P_c + f$, where f is the apparent slip fraction. Then the value of C is 450.

Example (i.).—What should be the diameter of the screw of a tramp steamer whose prismatic coefficient is 0·85, the I.H.P. 1,000, the speed is 10 knots, and the apparent slip to be 10 per cent.?

Here $$x = 0{\cdot}3 \times 0{\cdot}85 + 0{\cdot}10 = 0{\cdot}355.$$

$$P \times R = \frac{10 \times 6{,}080}{60 \times 0{\cdot}9} = 1{,}126.$$

$$D = \sqrt{\frac{1{,}000}{0{\cdot}253 - 0{\cdot}063} \times \left(\frac{450}{1{,}126}\right)^3} = 18{\cdot}3.$$

Example (ii.).—Determine the diameter of the twin screw of a ship whose P_c is 0·6, the I.H.P. 6,000, and the speed 20 knots, the apparent slip to be 15 per cent.

Here $$P \times R = \frac{20 \times 6{,}080}{60} \div 0{\cdot}85 = 2{,}384,$$

and $$x = 0{\cdot}2 \times 0{\cdot}6 + 0{\cdot}15, \text{ or } 0{\cdot}27.$$

$$\text{Diameter of screws} = \sqrt{\frac{3{,}000}{0{\cdot}27 - 0{\cdot}073} \times \left(\frac{450}{2{,}384}\right)^3} = 10{\cdot}1 \text{ feet.}$$

Example (iii.).—What should be the diameter of the screws of the triple-screw turbine-driven scout, whose $P_c = 0{\cdot}55$; her speed is to be 27 knots with 15,000 H.P., and the apparent slip 25 per cent.?

In this case $$x = 0{\cdot}18 \times 0{\cdot}55 + 0{\cdot}25, \text{ or } 0{\cdot}34 \text{ for the centre screw,}$$

and $$x = 0{\cdot}20 \times 0{\cdot}55 + 0{\cdot}25, \text{ or } 0{\cdot}36 \text{ for each wing screw.}$$

$$P \times R = \frac{27 \times 6{,}080}{60 \times 0{\cdot}75}, \text{ or } 3{,}648 \text{ feet.}$$

$$\text{Diameter of centre screw} = \sqrt{\frac{5{,}000}{0{\cdot}34 - 0{\cdot}116} \times \left(\frac{450}{3{,}648}\right)^3} = 6{\cdot}51.$$

$$\text{Diameter of wing screw} = \sqrt{\frac{5{,}000}{0{\cdot}36 - 0{\cdot}13} \times \left(\frac{450}{3{,}648}\right)^3} = 6{\cdot}4 \text{ feet.}$$

Example (iv.).—A four-screw ship of 60,000 H.P. has a P_c of 0·7; her speed is 25 knots, and the slip per cent. 16, of what diameter are the screws?

Here $$x = 0{\cdot}2 \times 0{\cdot}7 + 0{\cdot}16, \text{ or } 0{\cdot}30.$$

$$P \times R = \frac{25 \times 6{,}080}{60 \times 0{\cdot}84} = 3{,}017.$$

$$\text{Diameter of screw} = \sqrt{\frac{15{,}000}{0{\cdot}3 - 0{\cdot}09} \times \left(\frac{450}{3{,}017}\right)^3} = 15{\cdot}8 \text{ feet.}$$

If, however, the slip is to be only 12·5 per cent.—

$$x = 0{\cdot}14 + 0{\cdot}125 = 0{\cdot}265.$$

Then the diameter is 17 feet.

The Pitch of the Screw depends on the speed of the ship and amount of acceleration required to be given to the stream of water to produce the required thrust, or perhaps it should be said to produce the maximum amount of thrust possible with the shaft horse-power transmitted to it. That is to say, the pitch of screw is arrived at by dividing the sum of the speed of the ship and the "slip" in feet per minute by the revolutions.

The amount of slip required will depend on the size of the column of water projected, as in the case of the paddle-wheel (p. 464)—that is, as the disc area is decreased so that the column is diminished in volume the acceleration given it to produce the same thrust as before must be increased. Taking the same notation as before for speed and velocity, and A as the area of screw disc, its diameter D and its pitch P—

$$A = \frac{\pi D^2}{4} = 0{\cdot}7854\ D^2.$$

$$\text{Then momentum or thrust} = 2\,A \times V\,(V - v)\ \text{lbs.}$$
$$= 1{\cdot}57\ D^2 \times V\,(V - v)\ \text{lbs.}$$

Taking $(V - v)$ as $f\,V$ (a fraction of V). Then

$$\text{Thrust} = 1{\cdot}57\ D^2 \times f\,V^2\ \text{lbs.}$$

As for given speed and size of ship, etc., thrust may be assumed constant, then

$$f\,D^2 . V^2 = \text{constant.}$$

That is, $D \times V$ vary inversely as $\sqrt{f}$.

Thus, if the fraction of slip is increased, the value of $(D . V)^2$ decreases, and as $V = R \times P$, any variation in velocity V may be made by a variation in pitch or revolution, or both.

Pitch of Screw.—Let S be the speed of the ship in knots, R the number of revolutions per minute, and s the slip in knots; then

$$\text{Pitch} = \frac{(S + s) \times 6{,}080}{60 \times R}.$$

If the slip is expressed as so much per cent. of *the speed of screw,* which is the common way, and is x per cent. Then

$$S = \text{speed of screw}\left(1 - \frac{x}{100}\right),$$

or
$$\text{Speed of screw} = S \div \left(1 - \frac{x}{100}\right) = S \times \frac{100}{100 - x}.$$

Then
$$\text{Pitch} = \frac{S \times 100 \times 6{,}080}{60\,R\,(100 - x)} = \frac{S}{R} \times \frac{10{,}133}{100 - x}.$$

Example.—To find the pitch of the screw for a ship whose speed is 15 knots, the slip is 10 per cent., and the number of revolutions 60 per minute.

$$\text{Pitch} = \frac{15}{60} \times \frac{10{,}133}{100 - 10}, \text{ or } 28{\cdot}14 \text{ feet.}$$

The apparent slip of a well-proportioned screw on an ordinary cargo ship of fairly good form should be about 8 per cent. when at *sea* full speed, and 10 per cent. when on *trial* full speed. High-speed ships have higher rates—viz., from $12\frac{1}{2}$ to 16—with reciprocators and 15 to 25 with turbines. The large propellers of bluff cargo boats already alluded to, however, seldom exceed 5 per cent., and to them the remark does not apply. An excessive amount of slip does not of necessity imply waste of power, as it may arise from smallness of diameter; and also when a screw of larger diameter gives a better speed than that obtained by the original small one, it is often due to the increased efficiency of machinery at a reduced number of revolutions.

An abnormally large amount of slip may, however, be due to want of area of screw blade, and when this is the case the curve of indicated thrust exposes it, as it will then exhibit want of proper augmentation of thrust at the higher speeds of revolution.

The pitch of a screw should never be less than the diameter if it can be avoided, as in practice it seldom happens that such screws produce satisfactory results, although, since the turbine has come into use, designers have been compelled to supply screws whose pitch ratio is less than 1·0 to suit the revolutions necessary to high efficiency of the motor. It really means that the *angle of blade at tip* should not be less than that whose tangent is 0·32, or 18°.

Pitch Ratio is the ratio of the pitch of a screw to its diameter and an expression in common use and convenient in algebraic expressions. It is now sometimes as low as 0·75, but generally is over 1·0, and best results are obtained with values of 1·2 to 1·6. Screws are seldom found now with such high pitch ratios as once were common when high revolution was obtained by gearing to the slow-running engine. Such high-pitch ratios were also necessary and quite appropriate to vessels using large sail power with steam. It was no easy matter with such ships to keep the engines from racing badly —that is, from too rapid revolution during a squall.

Surface Ratio is also an equally popular and useful expression for the ratio of the actual blade area to that of a disc of equal diameter. It is quite as important that the surface of the blades shall be sufficient for the power transmitted or the thrust to be produced as that the diameter and pitch are right. It is claimed that by taking the total thrust and dividing it by the area of the blades as projected on a plane transverse to the axis the quotient should not exceed a certain figure depending on the immersion, otherwise cavitation is liable to ensue. The figure in question is said to be from 14 lbs. per square inch with shallow fast ships, such as scouts and destroyers, and 16 to 20 in the better immersed screws of the merchant ship. It is obvious that, with the small diameter screws of turbine steamers the surface ratio must be very much larger than that with similar ships of equal power with screws of larger diameter, although the actual surface of the latter may be the greater. Surface ratio, therefore, cannot be taken as any guide, except as a comparison with similarly driven ships.

The Acting Surface of the Blades of a Screw can be best determined by referring to successful practice as the guide, and experiments with models in tanks as a light. Prof. Taylor, U.S.A., as well as Dr. R. E. Froude, has done yeoman service for engineers in making such elaborate, careful, and exhaustive experiments, and publishing them, together with useful rules, etc., for the service of designers, etc.*

Screws of the same diameter may vary in form of blade in amount of

* Which may be studied with advantage in Prof. Biles' book on the *Design and Construction of Ships*, vol. ii., C. Griffin & Co

acting surface and in pitch; the first may be eliminated, inasmuch as modern screws do not differ much in that respect, there remains their surface ratio and pitch ratio as variants when making comparisons. With variations in pitch, however, there must be differences of slip ratio, involving questions of revolution. To examine the performance of a screw, there must be only one variable at a time, and the test or criterion of performance is the ratio of thrust to the torque producing it—that is,

E the efficiency = thrust $\times$ travel of ship $\div$ torque $\times$ 2 π $\times$ revolutions.

If, therefore, one screw may be examined by running it at different revolutions, a base line of revolutions may be taken, and with ordinates of the thrust and torque a pair of curves may be drawn, and with the various values of E a curve of efficiency. Also, a curve of slip may be constructed; or with a varying slip ratio as a base line curves of power efficiency can be constructed, and the results of the trials critically examined. The same screw may be tested with pitch ratio as a variant, and the power constant or the thrust constant and curves drawn with pitch ratios as the base or abscissæ. Another screw with a different surface ratio may be put through a similar set of curves constructed. Any one of these latter may be compared with the corresponding one of the former by means of their curves of efficiency. In this way it is possible to select the screw which shall best suit the conditions of the motor driving it by superimposing on the curves of efficiency that of the motor at various rates of revolution, and combining them so as to produce a new curve of general or combined efficiency.

Prof. Taylor made a long series of experiments with screws 16 inches diameter, the pitch ratio varying from 0·4 to 1·5 in six steps, and the surface ratio from 0·1207 to 0·5635 by five steps. He used as abscissæ the slip per cent., and constructed curves of efficiency and power for each of the pitch ratios of a particular screw. He also experimented with screws having two, three, and four blades, so that the surface ratio was varied by variation in number of blades, as well as by the breadth of them. It will be understood, then, that the number of such experiments was very great, especially as many of them had to be repeated; the figures involved enormous; the investigations of the results and their general application to practical engineering is a matter of special study, which may be made of them in Prof. Taylor's own book, or in that of Prof. Biles', which contains the chief part of them.

Prof. Taylor's investigations, however, were not limited to pitch ratios, slip ratios, and surface ratios; he has tested screws of different transverse sections, and shown that many of the old ideas respecting such were erroneous.

The Acting Surface of a Screw Propeller may be calculated by taking the power developed per revolution, and a factor which is governed by the fineness of the ship's water lines and the number of blades; it is also dependent on the position in the ship whether central or on the sides, as with twin screws.

RULE (1).—Area of acting surface in square feet = $K \sqrt{\frac{\text{I.H.P}}{R}}$.

Here $K = P_c \times M$, where

For four-bladed single screws,		M = 20;	for twin screws,	15·0.
,, three-bladed	,,	M = 19;	,,	14·3.
,, two-bladed	,,	M = 17·5;	,,	13·1.

The Thrust of a Screw may be deduced with a fair approximation to accuracy by the following formula, where A_s is the acting surface; P_r the pitch ratio; D the diameter in feet; V the velocity of screw in feet per second, or $\frac{P \times R}{60}$; G a coefficient depending on the disposition of the surface of the blades, and which varies in value from 0·42 of the mercantile ship to 0·5 of the broad-tipped small diameter screws with high-power turbine steamers.

$$\text{Thrust in pounds} = \frac{D \times \sqrt{A_s} \times V^2}{P_r} \times G.$$

For ordinary leaf-shaped blades, . . .	$G = 0·40$.
,, circular blades, ...	$G = 0·42$.
,, broad-tipped, as with turbines, . .	$G = 0·45$ to $0·50$.
,, shaped as generally found in merchant ship,	$G = 0·42$.

The Area of Acting Surface of a Screw may also be calculated by means of this formula for thrust.

From the above the following is deduced :—

$$\text{Area of acting surface} = \left(\frac{T \times P_r}{D \times V^2 \times G}\right)^2.$$

Taking the slip as a fraction of V, so that it is $f \times V$, then—

Speed of ship v is $V - fV$, or $V(1 - f)$.

The efficiency of engine and propeller is, as before, E ; then—

$$\text{Thrust in pounds} = \frac{\text{I.H.P.} \times 33{,}000 \times E}{60\,v} = \frac{\text{I.H.P.} \times 550 \times E}{V(1-)\,f}.$$

Substituting this value of f—

$$\text{Area of acting surface} = \left\{\frac{\text{I.H.P.} \times (550 \times E) \times P_r}{D \times V^3 (1-f)\,G}\right\}^2.$$

For the ordinary merchant cargo steamer, .	550 E = 330.
,, express and naval reciprocators, .	550 E = 360.
,, turbine direct-driven ship, . .	550 E = 380.

Example (i.).—What should be the surface of each screw of a twin-screw reciprocator ship of 6,000 I.H.P.; $R \times P = 2{,}400$; $P_r = 1·4$; and slip 20 per cent.; the diameter is 10 feet?

$$\text{Area} = \frac{360 \times 3{,}000 \times 1·4}{10 \times 40^3 (1 - 0·2)\,0·45} = 42·8 \text{ square feet.}$$

Example (ii.).—What surface should the screw of a tramp steamer have which has engines indicating 1,200 H.P. at 75 revolutions; it has to have four blades, and the prismatic coefficient is 0·85 ?

$$\text{Surface of blades} = 0·85 \times 20 \times \sqrt{\frac{1{,}200}{75}} = 68 \text{ square feet.}$$

Example (iii.).—A twin-screw steamer, whose prismatic coefficient is 0·6.

indicates 6,000 horse-power at 130 revolutions. The screws are to be three-bladed, and for smooth-water service.

Here $$\text{surface of blades} = 0{\cdot}6 \times 14{\cdot}3 \times \sqrt{\frac{3{,}000}{130}} = 41{\cdot}2.$$

Example (iv.).—A turbine-driven steamer crossing the Atlantic develops 24,000 S.H.P. at 250 revolutions; her prismatic coefficient is 0·66, and she has three screws dividing the power evenly.

$$\text{Here surface of blades} = 0{\cdot}66 \times 14{\cdot}3 \times \sqrt{\frac{8{,}000}{250}} = 53{\cdot}8 \text{ square feet.}$$

Thickness of Blade.—The strength of a blade at the root, or part near the boss, will vary nearly directly as the breadth and as the square of the thickness, and for practical purposes may be assumed to do so. The strain on the propeller will vary as the cube of the diameter of the tunnel shafts, and very nearly as the cube of the diameter of the propeller shaft. In similar engines, the twisting power of the engines has been shown to vary as I.H.P. ÷ revolutions.

Let d be the diameter of the screw shaft close to the propeller, b the breadth of blade in inches at a distance of $1\frac{1}{2} \times d$ from the centre, and n the number of blades; R the number of revolutions per minute. Then

$$\text{Thickness of blade (at } 1\tfrac{1}{2} \times d \text{ from centre)} = \sqrt{\frac{d^3}{n \times b} \times K},$$

$$\text{also} \quad ,, \quad ,, \quad ,, \quad ,, \quad = \sqrt{\frac{\text{I.H.P.} \times f}{R \times n \times b} \times K};$$

f may be taken as 100 for compound engines of the ordinary two-cylinder type; 90 for three-cylinder compound, four-crank triple, and quadruple engines; 85 for three-crank triple-compound engines, and 75 for turbines. The value of K is 4 for cast iron, 2 for ordinary gun-metal, 1·6 for cast steel, and 1·5 for forged steel and bronzes of superior make. For ships engaged in the North Atlantic or other seas where rough weather largely prevails, these factors may with advantage be increased by 20 to 25 per cent. This is especially necessary for cargo vessels, which may often have to run "light."

The thickness of metal at the tip should be 0·2 of that at the root.

Propellers made of steel and bronze are very superior in strength to those of cast iron, even if made in accordance with these rules; for if made of the *same* strength only, the blades would not be *stiff* enough, and would vibrate very considerably, especially when racing.

Fig. 173 is a typical design of solid cast-iron propeller. It is also typical of the form of the blade of a loose-bladed propeller, and the proportions are such as would be found generally with those having three blades. The Admiralty, as well as many private engineers, prefer to have the blades rounded at the top as shown, but blades in the past, as well as many of them in the present, are shaped as shown in dotted lines at M.

The surface of propellers designed approximately to fig. 173 may be found by the following rule:—

$$\text{Aggregate surface} = D \times B \times n \div F.$$

D being the diameter in feet, B the maximum breadth in feet, n the number of blades, and F for rounded tips, 3, and for square ended, 2·85.

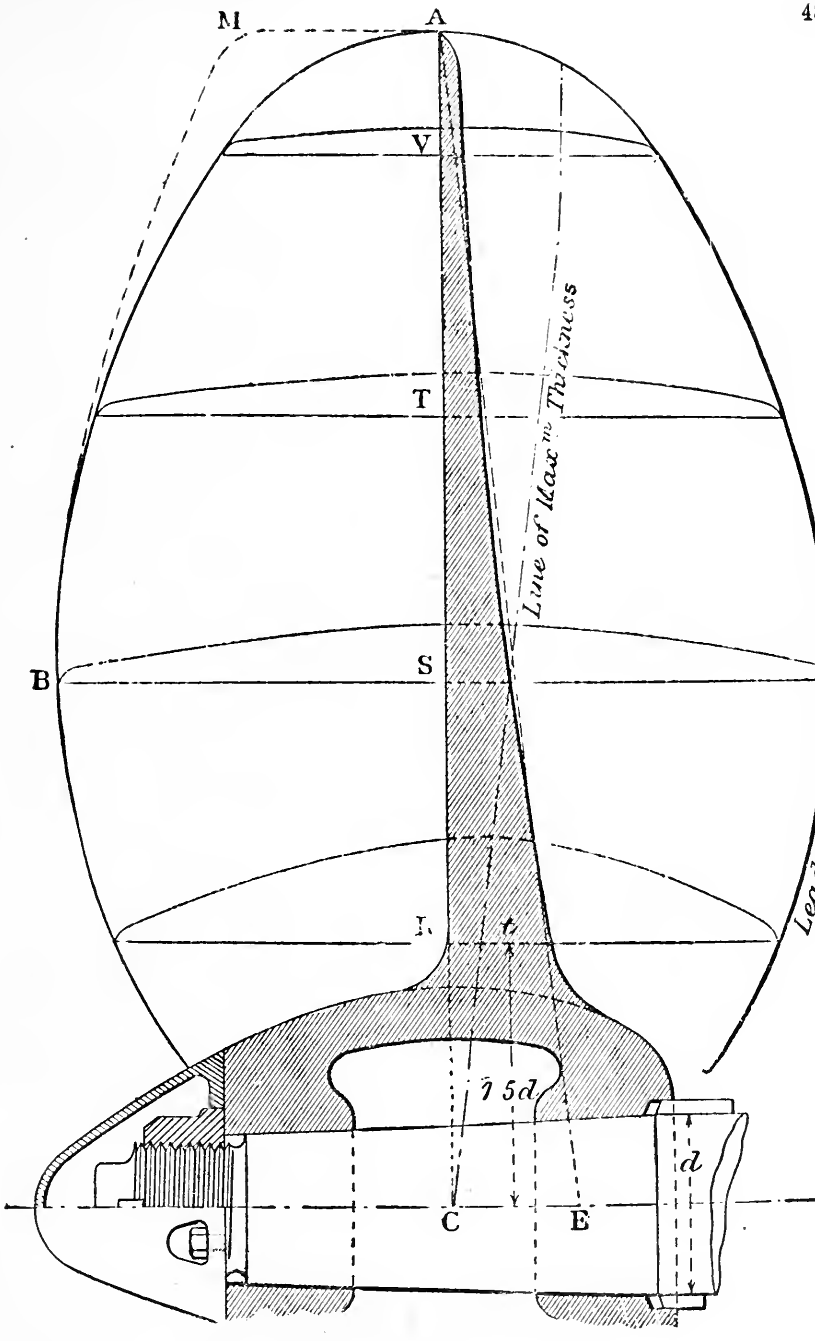


Fig 173.—Solid Cast-iron Screw Propeller.

The blade sections are obtained by taking a point E, so that

$$C E = \sqrt{\frac{d^3}{n \times b}} K^1,$$

d being the diameter of the screw shaft at the inner end or that of the tunnel shafting, n the number of blades, b their maximum breadth in inches. The value of K^1 is, for cast iron, 6·3; for ordinary gun-metal or bronze, 3·2; for cast steel, 2·5; and for bronzes of superior strength, 2·35.

Taking T as the thickness at $\frac{1}{9}$th of the diameter, or 1·5 d from the centre, the thickness at $\frac{2}{9}$ths should be 0·65 T. at $\frac{3}{9}$ths = 0·4 T, and at $\frac{5}{9}$ths = 0·23 T.

Propeller Boss.—When for a loose-bladed propeller, this is usually spherical in general form, with flats or recesses for the blades. The diameter of the sphere is from $\frac{1}{4}$ the diameter of the screw for small propellers, to $\frac{1}{5}$ the diameter for large ones, and may be taken at

$$= 0·9 \sqrt{\text{diameter screw}}.$$

The length of the boss depends on the size of the base of the blades, and is generally about 0·85 the diameter of sphere for a two-bladed screw, and 0·75 when four-bladed.

The bosses of some very large screws have been made oval in fore and aft section, and the base of the blades made oval to fit them. This form is not so good for many reasons as the spherical, and was adopted more on account of the boss being a forging than from any other cause.

In H.M. Navy, as well as in the navies of most countries, bronze of some kind is the material employed for both boss and blades. The Admiralty used to specify that it must be composed of 87·65 best selected copper, 8·32 tin, and 4·03 best Silesian spelter.* This mixture, when carefully made, should have an ultimate tensile strength of 16 tons per square inch, and be very tough. Unless very carefully melted, however, test pieces from large castings seldom exceed 14 tons.

The Admiralty have also used phosphor-bronze, Stone's and Parson's manganese-bronze propellers, which are more suitable to the service under present conditions. Bronze is a necessity with some naval ships, on account of the copper on the bottoms and the brass elsewhere in the submerged parts affecting cast iron or steel, but generally it is employed because its strength is superior to that of cast iron, and steel is viewed with some considerable amount of distrust, which has not been lessened by the action of some of the larger steamship companies, who have now generally adopted bronzes in lieu of steel. The zinc bronzes do not affect the hulls as tin bronzes did.

In the mercantile marine, the boss is generally of cast iron, of as strong a mixture as can be made, or of cast steel. In large steamers of full power, and all ships which make long voyages to distant parts, the bosses should be of cast steel, whether the blades be of iron, steel, or bronze.

The studs for securing the blades to the boss are of bronze for bronze blades, and sometimes for iron and steel blades, but for the latter best steel is preferable, the nuts being of brass with closed ends to protect the studs.

The bosses of solid cast propellers are oval in fore and aft section as a rule, and are from $\frac{1}{6}$ to $\frac{1}{8}$ the diameter of the screw in diameter, and about $2\frac{1}{2}$ to $2\frac{3}{4}$ times the diameter of the shaft in length.

* Of late years the Admiralty have decreased the amount of zinc (*v.* Chap. xxx.).

Screw Propeller Blades.—The screws of cargo ships of the mercantile marine are generally of cast iron, and up to moderate sizes are almost invariably cast solid, although generally there is a decided tendency to adopt those with loose blades. A solid screw is rather more efficient than one whose blades are bolted on, and is only about half the cost of the latter. The cause of inefficiency arises from the resistance of the projecting nuts, and often this is added to by the clumsy flanges of the blades, which project beyond the boss itself. A well-designed and carefully made screw should have the base of the blade conforming to the general outline of the boss, and the nuts or bolt heads recessed into the blade base, and covered in with a metal case or

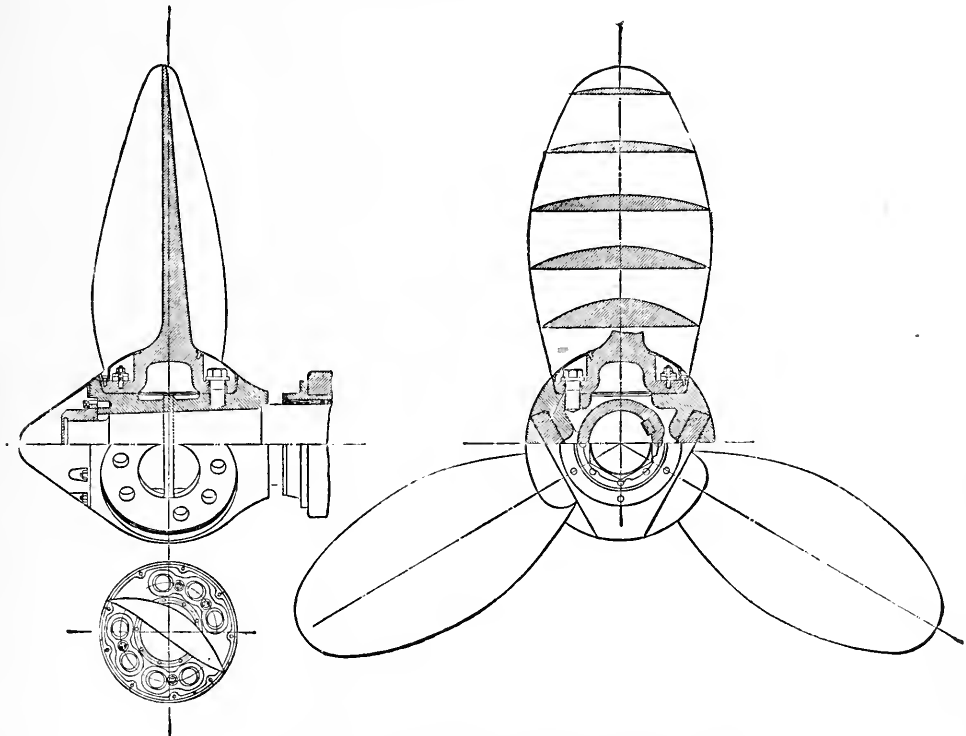


Fig. 174.—Screw Propeller Blades and Boss Bronze.

cement flush with the surface (fig. 174). The screws in H.M. Navy are generally made in this way, and are then quite as efficient as a solid screw.

If one of the blades of a solid screw is broken off, the whole screw is practically ruined,* and the expense of a new one thereby entailed; also, what is of more serious consequence, the ship must go into a dry dock, or on a slip or "hard" to have it removed, which operation is long and tedious, as the boss must be forced off, the tunnel-shaft taken down, and the screw-shaft withdrawn before the new one can be fitted. On the other hand, if the screw has its blades bolted on, and one is damaged, it can be removed by a

* A clever founder can burn-on new ends to blades both with iron and bronze.

diver, and a new one fitted in its place at a very small expense, and in a very short time; and even when an experienced diver cannot be obtained, the ship can be tipped by discharging cargo from aft, etc., or shifting it forward.

The forms of propeller blades are so numerous and varied as to be beyond description here. The well-known blade (fig. 180) introduced by Mr. Griffiths, and now bearing his name, was the one most generally adopted by engineers, and generally gave satisfaction. Only a few carry out the plan he usually so strongly recommended, of bending the blade slightly forward, perhaps principally because it then came too near to the stern-post. Some have tried blades bent in the reverse way, and satisfied themselves that improved performance is obtained. The data published by inventors of screws are generally very misleading, as the failures are never mentioned, and seldom is it observable that an old screw has been replaced by a patent one of the same diameter, pitch, and area. There is little doubt that the better results with the "improved screw" are due rather to better proportions than to the particular *shape* of blade. The number of patents relating solely to the *form* of blade is endless, and some special ones, introducing additions "to prevent loss from the centrifugal action," reappear periodically as novelties. This latter is a great bugbear with many engineers; for after all, the loss from this cause is non-existent, while it always happens that means adopted to prevent it are themselves causes of a considerable loss from frictional resistance.

As a rule, the greatest breadth of blade should be beyond one-third of the diameter of the disc from the centre, and should be approximately as given by the following rule:—

$$\text{Maximum breadth of blade in inches} = K\sqrt[3]{\frac{\text{I.H.P.}}{\text{revolutions}}}.$$

For a four-bladed screw, $K = 14$; for three-bladed, $K = 17$; and for two-bladed, $K = 22$. For modern high-speed screws with turbines K is higher.

The breadth of blade at the tip should be from $\frac{1}{3}$ to $\frac{2}{3}$ the maximum.

Propellers are sometimes made with the blades of finer pitch near the boss than at the tip, partly that the angle shall not be so coarse that this part of the blade only churns the water, and partly that the hold on the boss due to the increased breadth of the blade may be greater. A decrease of pitch of 10 per cent. has given good results, and when the propeller is of small diameter, with a very coarse pitch, as much as 15 per cent. decrease may be adopted; as a rule, however, a true helix gives best propelling results, especially if the tail part of the sections is rounded off ship shape.

The Number of Blades is arbitrary; the designer may have two, three, four, and even six, for experiments both with models and real screws have shown that the efficiency is not seriously effected by the number. It is, however, certain that in smooth water, and neglecting vibration as a factor, the two- or even the one-bladed screw can show the highest efficiency, and that three is better for twin-screw ships of all kinds; but for rough water with moderate immersion four blades are found to be generally most effective in large single- and twin-screw ships. The efficiency of propellers with a larger number of blades is not so good as with four, while the weight and cost is greater. In practice the four-bladed screw is universally employed in the deep-water mercantile marine, and three-bladed ones in the Navy and for smooth-water express steamers.

It remains for future experimental trials with turbine-driven ships to decide whether the necessary surface is better distributed over five or even six moderately wide blades than over three or four with abnormally wide tips. The experiments made by the Admiralty with six-bladed screws in

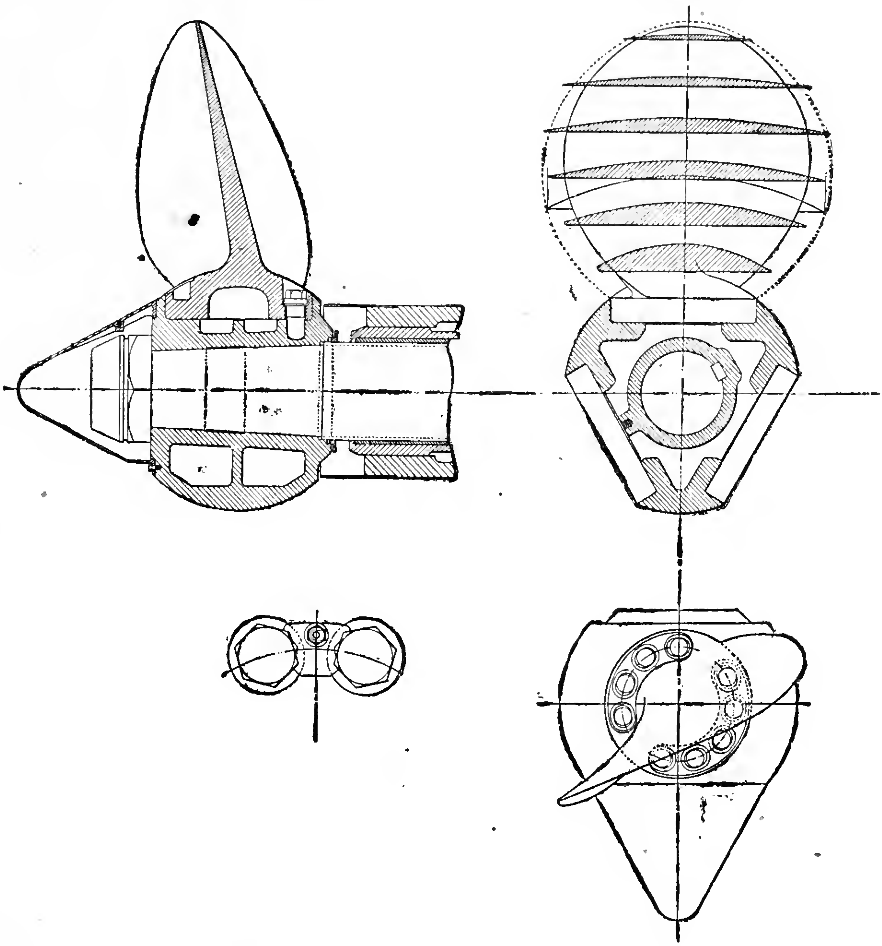


Fig. 175.—Modern Fast-running Bronze Screw for Cruiser.

H.M.S. "Shannon" showed a decrease in efficiency of 5·6 per cent. with the six as against four blades; but in that case the surface was proportional to the number of blades, and, therefore, excessive with the six. A six-bladed screw in H.M.S. "Emerald," having a surface only 4 per cent. greater than that of the four-bladed of the same diameter and pitch, gave an indicated thrust of only 6 per cent. less than did the four-bladed. But the speed

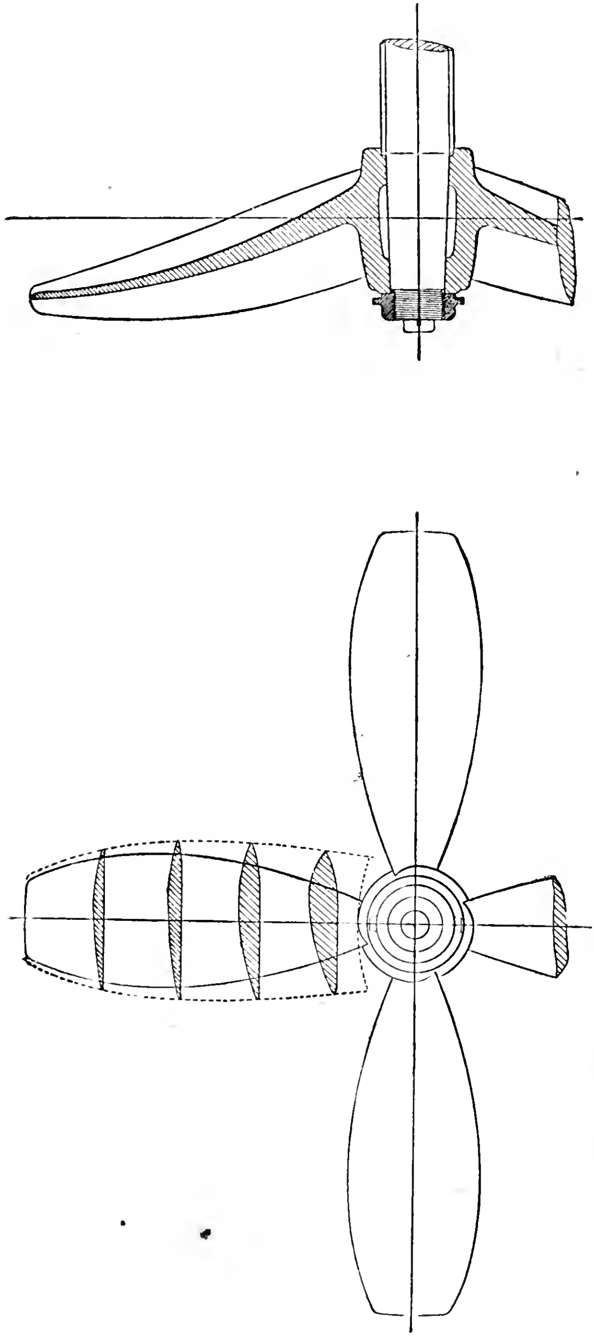


Fig. 176.—Mercantile Service Screw.

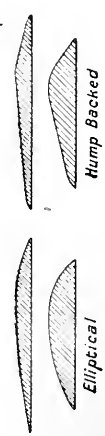


Fig. 176a.

coefficient with the six-bladed screw was 175·1 against 171·7 with the four-bladed, and the slip per cent. 11·26 against 12·28; the speed, however, was 12·003 with the four-bladed screw as against 11·726 with the six, and on this the case appears to have been decided in favour of the former.

The Shape of Propeller Blades approximate more or less to that introduced by Mr. Griffiths, and used in the Navy from 1860, as shown by the dotted line in fig. 173. To-day naval screws, as well as a large number of those

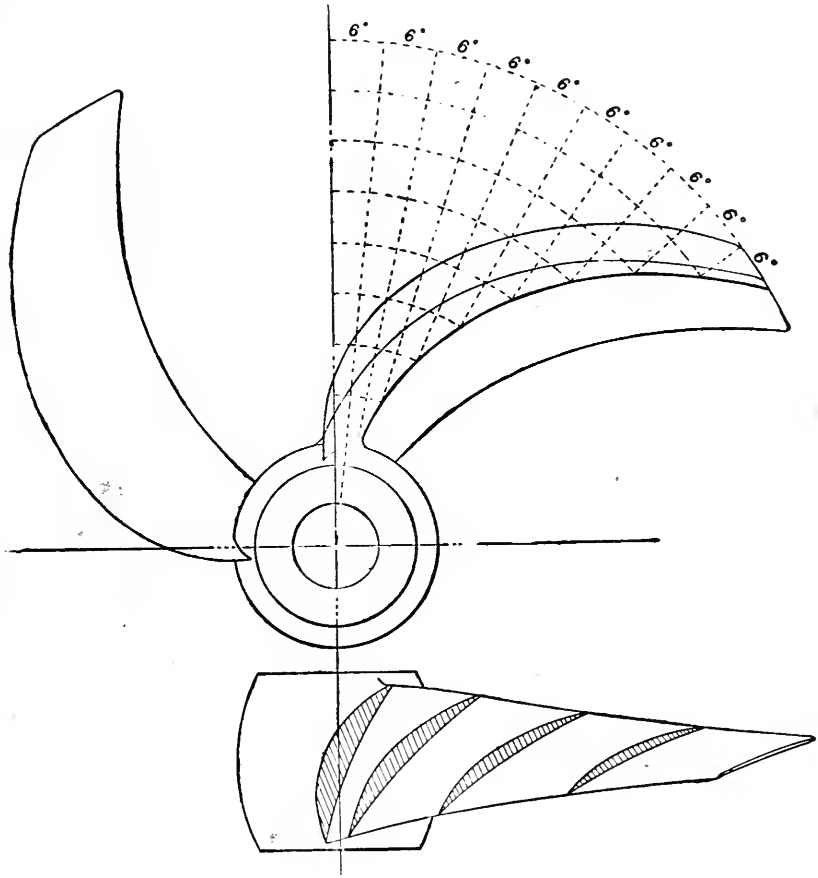


Fig. 177.—Hirsch's Screw.

in the mercantile marine, are shaped as shown in the full lines of that figure, but they generally are made with loose or fitted blades, as shown in fig. 174. With the increase in rate of revolution in the full-powered high-speed twin-screw express steamer and cruiser, the surface ratio has been largely increased, so that the blades have assumed almost a circular form, as shown in fig. 175. While the demand for surface for the very small diameter screws of the turbine-driven ship has caused a further increase in the breadth of blade, so as to approximate to an ellipse with the minor axis radial, the surface ratio in their case is very high, even as much as 0·56 to 0·60. In the ordinary cargo boat with its four-bladed screw of large diameter the surface ratio is only about 0·37, and the blades generally retain the squareness of tip as formerly obtained, inasmuch as the outer part of the blade is the most effective, especially when running in ballast trim. Fig. 176 is a good example of such a screw with the blades inclined so as to be as far removed from the body of ship as possible, and thereby work in better water.

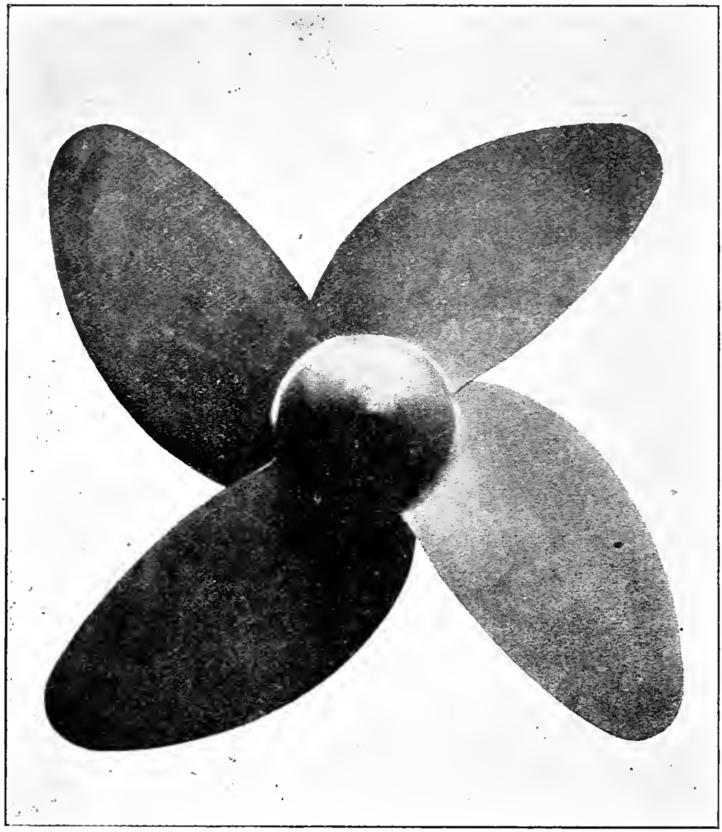


Fig. 178.—One of the new Propellers of R.M.S. "Mauretania," 68,000 S.H.P.

(Of "Parsons" Special Turbadium Alloy.)

There have been from time to time new forms of screw which have established a reputation for a while, but not finally survived. One of the most noteworthy of these, *Hirsch's Patent* 60° *Screw*, as shown in fig. 177, was a favourite one, both on the Continent and in Great Britain, and deservedly so, inasmuch as it was quite successful in competition with even good screws, and when replacing inferior ones marked improvement in propelling efficiency was always obtained; but it was not superior to the best designs with the ordinary leaf-shaped blade, as in fig. 174, of true pitch. As a stern-going propeller, Hirsch's screw was very successful, and at the time it was tried vibration with it was less than with the screws then obtaining.

The Section of Screw Blades is usually an approximation to a segment of a circle, as shown in fig. 175, but it is the fashion with some engineers, who claim for their screws a higher efficiency thereby, to make them of a transverse section, shown in fig. 176*a*, with either the hump-back or the rounded one. In both cases, what may be called the "forebody" is not so fine as the after or following portion. Experiments by Prof. Taylor have proved what had been for some time a surmise—viz., that the blade with a ship-shape cross-section moves with the least resistance and highest efficiency. When the stereotyped segmental form near the boss is departed from, and the sections there become ship-shaped, there is always an improvement, especially at high speed of revolution. The mere rounding of the tail or following portion of the blade sections seems to influence the efficiency more than a similar modification with the leading half does. The effect of such changes, however, is virtually to decrease the pitch, especially near the root.

The Studs or Bolts of a Screw Blade should be of the strongest material, especially in these days of high rate of revolution, when the centrifugal forces which they have to resist are very high. Under ordinary working conditions the bolts on the side of the acting face are in tension, as they tend to keep the blade from tipping under the thrust on the surface; those on the back exert no action on them, unless the blades are simply attached to the boss without being recessed, when they are subject to sheer from the same pressure. But all the bolts help to hold the blade against centrifugal force, which is expressed by $\frac{W}{g} \times \frac{v^2}{r}$, where W is the weight in pounds of the blade, v the velocity in feet per second of the centre of gravity whose distance is r feet from the axis, and g, gravity = 32.

That is, Tension on the bolts due to C. Force $= \frac{W v^2}{32\, r}$.

If R be the revolutions per minute, and pitch is P, then

$$v = \frac{R}{60} \sqrt{P^2 + (2\,\pi\, r)^2}.$$

The Diameter of the Blade Bolts, when made of high tensile bronze or steel, whose number is n, can be found from the following:—

$$\text{Diameter of bolts or studs} = \frac{d \times Z}{n},$$

where d is the diameter of the solid shaft of sufficient size for the torque, and Z is a factor which, for a three-bladed screw, is 1·6, and for a four-bladed 1·3. The sizes obtained in this way are generally sufficient, but for high rate of revolution should be checked by estimating the tension on them due to centrifugal force, and allowing a stress not exceeding 4 tons per square inch of section at bottom of thread.

Material for Screw Blades.—Cast iron is, of course, the cheapest material for this purpose, and also possesses another advantage not so much appreciated until experience teaches—viz., that when struck violently against an obstacle like a jetty or wreckage, it breaks clean off, without, as a rule, damaging the shaft; on the other hand, steel or bronze blades, which are strong enough and tough enough to resist fracture, are sometimes bent so as to prevent the central screw from turning, and may cause the shaft to be bent or even broken (*v.* fig. 179). Cast iron when used should be of the very best description, twice cast and cooled slowly; hematite, and even steel, is often added to strengthen the mixture; the former is now generally used by moulders for this purpose, and has a tensile strength of 15 tons.

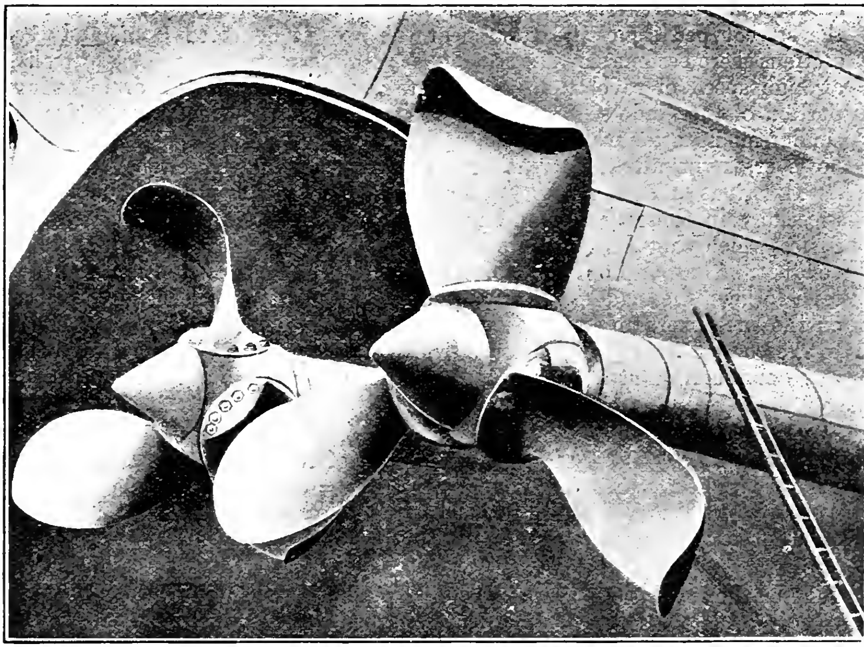


Fig. 179.—Damaged Bronze Screws.

Steel blades could be bought at about two and a-half times the cost of cast-iron ones; they are very much stronger, even when the section is considerably reduced, and consequently were more reliable, especially for engines of very large power, which must work at high speeds in rough weather. The efficiency of the propeller would be somewhat increased in consequence of the reduction in thickness, but steel blades were seldom true to pitch: hence when the best possible results are necessary blades of the strong bronzes should be fitted with their surfaces ground smooth and made true to pitch.

The chief objection to steel, especially to cast steel, is its liability to rapid corrosion on the backs of the blades. Cast iron corrodes into pits, but steel goes much more rapidly, and in some instances the tips are "honeycombed" in a few weeks. Corrosion, however, is not unknown with some of the bronzes, especially with screws of high-revolution; *turbadium*, however, and some other alloys are free from this defect as well as from erosion (*v.* Chap. xxx.).

Paint provides little or no protection, and even nickel plating, which has been resorted to on steel, has not proved very efficacious. Muntz metal sheathing has been tried, but with not very satisfactory results.

Phosphor-bronze, Parsons' special bronze, Bull's metal, Stone's bronze, Delta metal, etc., have taken the place of steel in most quarters, notwithstanding that the cost is about twelve times that of cast iron. Blades made of these materials can be cast very thin—thinner, in most cases, than if made of steel—because there is no loss of strength by corrosion; in fact, the very high speed of certain ships is attributed largely to the high efficiency of the propeller from the thinness and truth of the blades. Fig. 178 is a notable example of this in a huge solid screw.

Bronze propellers are objected to on the ground that injury is often done to the iron work near them by galvanic action; but the Admiralty does not hesitate to continue the practice of fitting them, nor do the large steamship companies. Corrosion does sometimes take place in such a way as to cause the source of the evil to be attributed to bronze screws; but the fitting of some slabs of zinc in way of the propeller prevents it, and removes all fear of ill consequences. The zinc bronzes are less liable to cause corrosion.

Weight of Screw Propellers may be calculated with a close approximation to truth by the following rule:—

$$\text{Weight in cwt.} = \frac{\text{Surface} \times \text{thickness}}{\text{K}}.$$

The surface is taken in square feet, the thickness in inches at the root of the blade.

K is for solid	cast-iron	screws,	4·5
,, ,,	bronze	,,	3·8
,, built	cast-iron	,,	3·0
,, ,,	cast-steel	,,	2·8
,, ,,	bronze	,,	2·5

Feathering Screws.—Yachts and ships which are required to sail as well as steam cannot well do the former when the screw is stopped, unless some means be adopted of feathering the blades, so that they are nearly in a fore and aft plane, or else by withdrawing the propeller altogether from the water. The late Bennett Woodcroft patented, in 1844, a plan for feathering the blades, which in a modified form was fitted by Messrs. Maudslay, Sons & Field to several ships. The blades, of which there are two, have shanks fitting into the boss, to which short levers are secured inside the boss; these levers are connected by links to a sliding collar outside the boss, which is carried round with the shaft, but is capable of being moved "fore and aft" on it by means of a pair of bell-crank levers actuated by a screw from on deck. When it was desired to sail, the blades were moved round into the fore and aft position by sliding the collar from the boss.

Bevis' Patent Feathering Screw.—Many patents were taken out for methods of effecting a similar movement of the blades, without the objectionable feature of external bell cranks, and to the late Mr. Bevis is due the perfecting of this idea. In 1858 Gregory and Craymer patented a feathering screw of which "the screw propeller shaft is made hollow, and a second shaft goes through it, carrying worm threads, which act on the propeller blades, so as to feather them to the angle desirable." In 1866 H. B. Young patented a

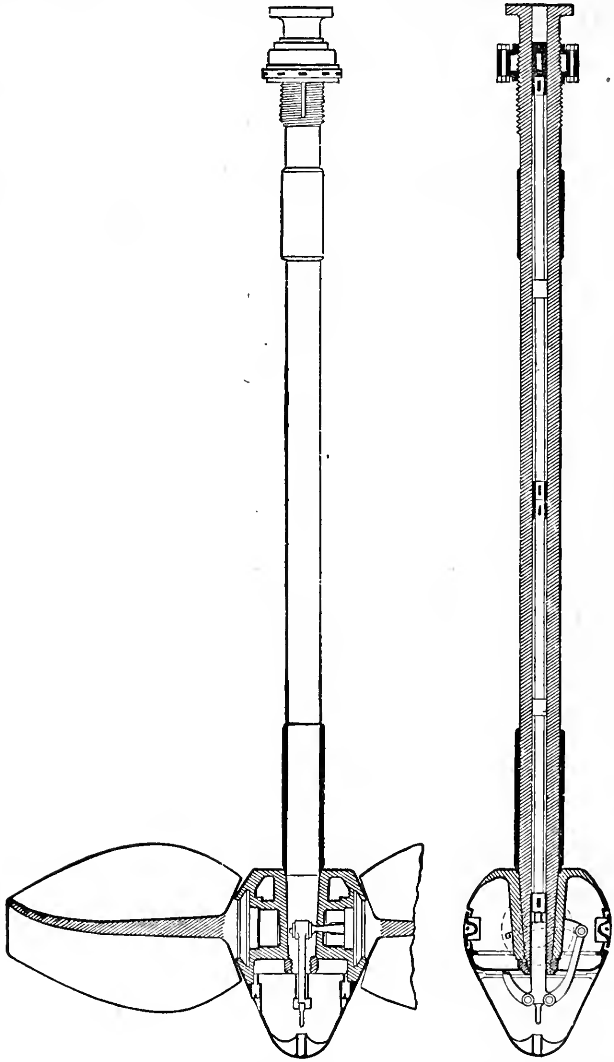


Fig. 180.—Bevis' Patent Feathering Screw.

somewhat similar idea as to the hollow shaft, but says "levers may be attached to the shanks." Fig. 180 shows the Bevis plan, which needs no description, and answers its purpose admirably.

The feathering screw can be converted into a *reversing* one by moving the blades through a greater angle, and, although their efficiency in that state will be very low, it will be good enough for casual astern going with a non-reversible oil engine of moderate power. It would be quite a satisfactory way of reversing small craft, and it will avoid the use of wheel gearing. *

Lifting Screws.—The screws of warships were formerly nearly always made and fitted, so that when desired they could be raised to the level of the deck for examination and repair when necessary, and to prevent obstruction when sailing. This plan is a very costly one, and not so efficient as the feathering blade, inasmuch as the ship steers badly owing to the gap in the deadwood, but it admits of examination and repair, which is of the utmost importance in a warship. A hole, however, is also necessary through the stern to admit of the screw coming on deck, which very much weakens what is already a somewhat weak part of the hull.

The lifting screw has a short piece of shaft cast with the boss or fitted to it in the usual way; the forward end of this shaft is provided with a driving piece, which is so formed as to fit into a slot across the cheese coupling keyed to the outer end of the stern shaft. The propeller is carried in a frame, called the *banjo frame*, which is arranged to slide up and down in grooved metal guides secured to the stern and rudder posts, and supported, when in working position, by two strong brackets or chairs also secured to these posts, and held down by two strong wooden sampson posts, whose upper ends are fitted with jam screws to the metal guides; such an arrangement is made wholly of gun-metal.

Number of Screws on a ship depends on various circumstances. Multiplication arose first with ships of light draught and good speed, inasmuch as before Yarrow and Thornycroft's discoveries it was the rule to have the screw immersed when at rest. *Twin screws* were found to have the advantage of safety and power to manœuvre. *A screw at each end* was the invention of James Howden for tugs, to provide for one being immersed when pitching, as such short boats do when towing. Three screws were used in foreign naval ships to permit of more elastic subdivision of power than obtains with twins. *Three screws* also were found an advantage with turbine direct-driven ships of large power, and with greater power still *four scerws* were necessary for the draught of water possible, and in naval ships for the subdivision of power desirable when cruising and manœuvring. *With the reciprocators and turbine combination* three screws obtain.

* This method of reversing is coming into use with ships driven by oil engines of moderate power. The screw blades are, of course, formed so as to be fairly efficient when in "astern gear."

CHAPTER XIX.

SEA-COCKS AND VALVES.

The importance of having efficient connections to the skin of the ship for inlet and outlet purpose has long been recognised, since neglect in the design or working of these fittings has caused damage, and sometimes the loss of a ship. Next to absolute safety, simplicity should be most aimed at in all parts of the machinery of a ship; but in no place, perhaps, is there greater need of careful consideration than in the arranging and designing of the sea and bilge fittings, so that neither a careless nor an ignorant engineer can endanger the safety of the ship; it must be "fool proof."

Lloyd's Committee have issued special rules on this subj ct, and the Board of Trade are no less vigilant in endeavouring to eliminate every source of danger, as indeed are all the other Register Societies.

It is needless here to give any illustration of how easily a ship may be flooded by an ill-considered arrangement of sea-cocks in ignorant hands, nor to give examples of ships that have foundered, for every engineer has heard of them. But even good arrangements may cause mischief, if, from want of simplicity, they are ill understood.

In the first place, the cocks and valves themselves should be of simple construction, and carefully marked, so as to be easily seen whether they are open or shut. The parts exposed should be either strongly made or carefully guarded, so that they are not liable to injury or derangement; and, when possible, so designed that they can be packed and examined without having to dock the ship. It is also necessary that the very large openings should have a second valve, as a safeguard in case of accident to the first one.

Sea-cocks and valves should be so placed that they are accessible at all times; and, as far as possible, within sight from the working platforms.

Kingston Valve.—For all large inlets the *Kingston valve* is preferable, as it acts as a non-return valve in case of the spindle breaking, and can then always be worked by simply forcing it outwards, either with the spindle itself, or by a rod substituted for it.

The Kingston valve is usually made in the form of a frustrum of a cone, with a taper of about 8 in 12; the length of the seat is generally about $1 \text{ inch} + \frac{\text{diameter}}{12}$. The object of this form of seat is to allow the valve to close itself tight in place, in case of the spindle breaking, by the pressure of the water. The Admiralty require that these valves shall be made in one with their spindles, of best gun-metal, and the spindle *tested* by a load of half a ton for each square inch of section of valve. For example, a valve 4 inches diameter, having an area of 12·56 square inches, should have a spindle sufficiently large to be tested to $6\frac{1}{4}$ tons.

Hence, if the proof stress of good gun-metal be taken at 6 tons (which is quite high enough) per square inch, then,

$$\left.\begin{array}{l}\text{Diameter of Kingston valve spindle at bottom}\\ \text{of thread,}\quad . \quad . \quad . \quad .\end{array}\right\} = \frac{\text{diameter of valve}}{3{\cdot}46};$$

so that a 4-inch valve should have a spindle 1·15 inch diameter at its smallest section.

But the Admiralty do not require any valve to be tested above 12 tons; so that for mere test purposes, no spindle need have more than 2 square inches in section, or be more than $1\frac{5}{8}$ inch diameter at its smallest part; but since for very large valves a spindle of this size would not be stiff enough, the following rule for all valves above $5\frac{1}{2}$ inches diameter holds good:—

$$\text{Diameter of spindle} = 1\tfrac{5}{8}\text{ inch} + \frac{\text{diameter in inches} - 5\frac{1}{2}\text{ inches.}}{16}$$

Kingston valve-boxes and tubes are generally made of gun-metal; but this is not a necessity, except where hot water or steam is blown through them, when cast iron would be dangerous. If the body is of cast iron, of course the valve seat should be of brass, and the working parts bushed with brass.

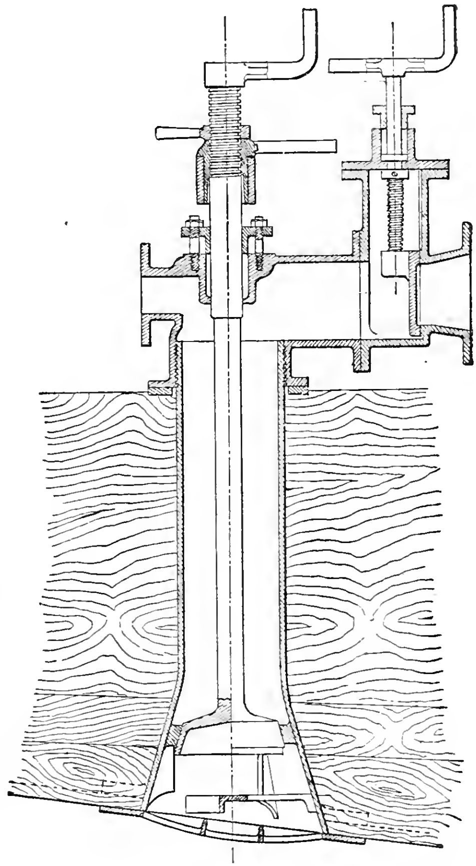


Fig. 181.—Kingston Valve (in a Wooden Ship).

Fig. 181 shows a good arrangement of Kingston valve for large sizes. It has a lifting nut secured in a bridge, as well as a handle on the spindle end; the former is used to start the valve or jam it in its seat, and the latter merely to open or shut it; the lock nut with handle is to secure it in any required position. In this figure the head is shown screwed on to the tube, and is such as was necessary in wooden or composite ships; but when fitted to the skin of an iron ship, a flange is formed at the bottom of the conical part, as shown by dotted lines and in Fig. 182 (A, B, C, D). A much simpler and less expensive plan is to form the valve like an ordinary stop-valve with four wings, and the spindle inverted—that is, on the same side as the wings; the bottom part of the box is then only slightly conical, and much shorter —in fact, only sufficiently long to allow of the valve lifting a distance equal to one quarter of its diameter, and in this position leaving space between it and the grating for the free flow of water.

All inlet valves should be fitted with a brass grating, whose meshes should not exceed half an inch in breadth, the total area through them being at least 20 per cent. larger than the net area of the tube. In the Navy, Kingston valves are fitted to all inlets and blow-off pipes, and are always supplemented with a cock or valve attached to them.

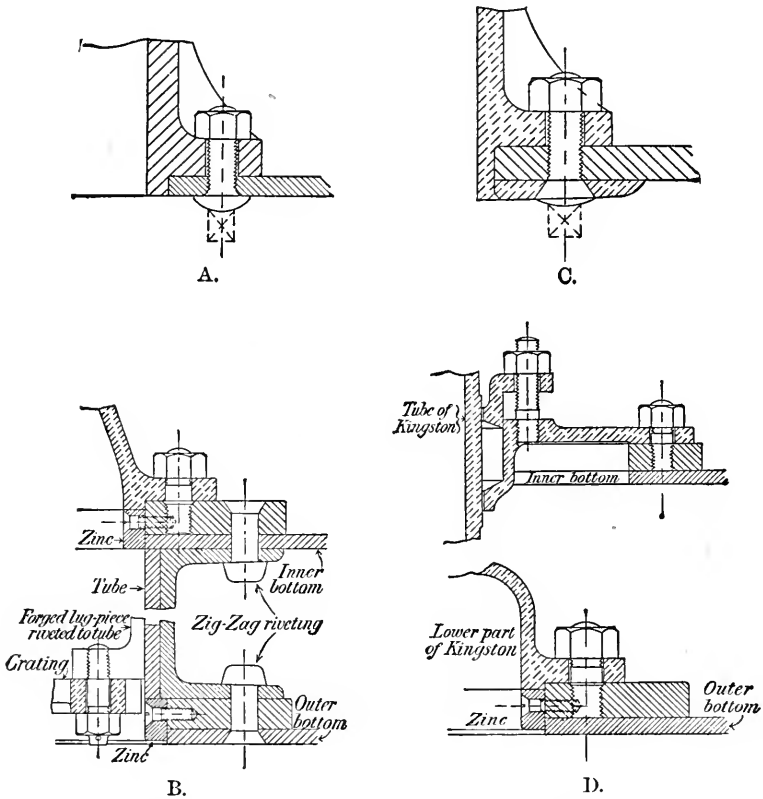


Fig. 182.—Details of Inlet Valves (in Steel Ships).

Valves are fast taking the place of cocks for all general purposes, so that whenever convenient a valve may be fitted in lieu of a cock to even the smallest Kingston, but to large Kingstons a cock could not be fitted, and so it is usual to find an ordinary sluice valve. The spindles of these supplementary valves should always be within easy reach of the platforms, and in case of the very large ones, when possible, they should be carried so high as to admit of the valve being worked when the engine-room is flooded.

In the merchant service, whether valves or cocks are fitted to the skin of the ship direct, the same precautions should be taken as enumerated above, and extra care taken to protect them from injury, as they are very liable to damage from the flooring plates washing about when a considerable quantity of water is in the bilge and the ship rolling heavily.

For the larger inlets of a merchant ship an ordinary stop-valve, opening inwards, is usually fitted, the box being of cast iron. A good plan, so as to avoid a number of openings in the ship's skin, is to fit a single box with one stop-valve in it, and that by preference an inverted one, and to this box fix the various cocks or valves necessary for the different requirements. For ships destined for a cold climate, a small cock can be fitted near the bottom of the box to admit steam to thaw any ice that may be blocking the orifice, it is also useful to blow away seaweed, fish, etc., which may be choking the meshes, and may be fitted with advantage to every ship.

Discharge Valves should be fitted to all outlets through the ship's side, and they consist, as a rule, of simple non-return valves having spindles passing through the covers, so that the valves may be lifted or pressed down as required. Here again a number of holes through the skin of the ship may be avoided by connecting the smaller valves to the box of a large one above the valve itself. In many cases this method is a very good one, having advantages beyond that already named—viz., that the pipes can be shorter, and the valves standing clear of the frames of the ship are easy of access.

The Board of Trade* require that all inlet and discharge valves shall be connected directly to the skin of the ship, and have no pipe or joint intervening (this rule, however, does not apply to the case of the smaller valves when attached to the box of a large one), and also that all discharge valves shall be placed above the load water-line.

The Admiralty, on the other hand, for obvious reasons, require all the discharge orifices to be below the water-line; and while in the merchant service the discharge valve-boxes are nearly always of cast iron, in the Navy they are invariably of bronze.

It is not unusual to fit the smaller discharge valves as simple non-return valves having no external spindle whatever; but this has the disadvantage that one is never certain if the valve is shut, and, if not shut, without the spindle it is beyond control.

It is sometimes found convenient to fit, for large discharges, a *straight-through* valve in lieu of the ordinary mushroom-valve. This straight-through valve is an ordinary flap valve resting on a vertical or nearly vertical seat, and lifted by means of a horizontal spindle, which passes through a stuffing in the side of the valve-box, having inside a slotted lever connected by a pin to a pair of lugs on the back of the valve, and, outside, a hand lever for controlling the valve.

* The Board of Trade regulations referring to sea connections are as follows:—

137. All inlets or outlets in the bottom or side of a vessel near to, at, or below the deep-load water line other than the outlets of water-closets, soil, scupper, lavatory, and urinal pipes, should have cocks or valves fitted between the pipes and the ship's side or bottom; such cocks or valves should be attached to the skin of the ship, and be so arranged that they can be easily and expeditiously opened or closed at any time; and the cocks, valves, and the whole length of the pipes should be accessible at all times.

Cocks or valves standing exceptional distances from the ship's plating, that is where the necks are longer than is necessary for making the joint, should not be passed without the sanction of the Board of Trade, and one condition of their being passed is that they should be made of gun-metal and well bracketed.

Bilge Valves.—As has been already said, too much care cannot be devoted to avoid all risk of flooding the ship by carelessness. An easy method of overcoming the difficulty in the case of pipes not greater than 4 inches diameter, is by means of the switch or three-way cock—that is, a cock having two side branches and a hollow plug with one orifice in the side and the other in the bottom, so that all water must pass by way of the bottom and one side only at one time. Suppose that an auxiliary engine is required to draw water from the bilge as well as from the sea, a three-way cock should be fitted so that its bottom is connected to the pump suction, one side branch connected to the sea inlet, and the other side branch to the bilge piping; it will then be easily seen that the two branches cannot be connected by the plug, and consequently, however careless the engineer may be, water cannot pass from the sea to the bilges. When, owing to the largeness of the piping, a cock cannot be fitted, self-closing non-return valves should be connected to the tail pipes leading to the bilge.

Communication Boxes.—All the bilge suctions, and all ballast-tank suctions, should be led to a *communication box* placed in some convenient position in the engine-room, above the level of the flooring, so that the requisite changes of service of pumping may be easily and quickly effected.

The communication box (called sometimes a "directing" box) consists of a rectangular chest having as many valves in it as there are pumping stations or parts from which water is to be drawn; under each valve is a separate tail, to which the suction pipes are attached; the valves themselves are of the ordinary mushroom type, non-return, and arranged so that each may be shut close by screwing down a spindle on it. The box cover is fitted by hinge bolts, so as to be easily and quickly opened for examination, and the joint made again.

To the upper part of the box the suction pipes of the pumps are fitted, so that the auxiliary pumping engines and engine bilge-pumps may draw from the same set of pipes. Both bilge-pumps should not be connected direct to one box, as it not unfrequently happens that it is required to pump water separately from two compartments at the same time, or that one of the bilge-pumps may be required when no sanitary pump is fitted to deliver on deck. The lid of the communication box should have inscribed legibly on it the lead of each valve.

Bilge Suction Piping is sometimes of cast iron and sometimes of lead, but the tails leading directly into the bilge should be of iron; cast iron is now often used instead of lead as less liable to damage, but it cannot be so easily cut in case of necessity. The Admiralty require all tail pipes to be of galvanised wrought iron; and all copper piping laid in the bilges to be covered with waterproof canvas varnished over, and otherwise insulated to avoid deleterious effects on the ironwork from galvanic action. All suction pipes to the bilge should fit into rose-boxes of iron-galvanised, and having an aggregate area through the holes at least twice that of the pipe. The cover of the box should be hinged, or so fitted that it can be opened and shut again very quickly. The boxes should also be so placed as to be easy of access. Bilge pump suction pipes should also be fitted with mud traps, consisting simply of a cast-iron box with a strainer and lid, in which the slight cessation of flow allows the deposit of the heavy dirt carried on so far with the water. All copper or brass-work subject to the action of bilge-water, should have the connections made with *bronze* bolts and nuts. Iron quickly corrodes unless well protected; and Muntz metal has been found to decay in a very peculiar way. Naval brass, however, which is Muntz metal improved with a little tin, is suitable for this purpose. Every care should be taken to protect the metallic surfaces from the corrosive action of bilge-water; for this purpose a good coat of Portland cement wash answers even better than paint or varnish.

When it is required to pass pipes, especially copper ones, through a water-tight bulkhead, a good plan is to fit a short length of cast brass, having a flange at each end to connect to the pipes on either side, and a collar in the middle, about 4 inches larger in diameter than the flanges, to secure it to the bulkhead.

If pipes are likely to expand considerably, or to be in such a position that the working of the ship in a sea-way would affect them, it is better to pass them through stuffing-boxes in the bulkhead. Very small pipes can be connected by union nuts to the bulkhead joint above described, thereby decreasing the size of the hole to be cut and the collar for closing it.

The Board of Trade require that one of the bilge-pumps shall be so fitted that it can pump water on deck in case of fire.

The engine-room pumps should have the necessary piping, valves, etc., that water may be drawn from each hold or compartment separately, from each side of the engine and boiler rooms, and from any other part of the ship liable to leakage or lodgment of water.

In addition to the above-mentioned means of freeing the ship from water, it is usual to have a bilge suction, so that the circulating pump may be utilised. The chief danger to be apprehended when the circulating pump draws from the bilge is, that in cases of flooding of the boiler compartment, the small coal is washed out of the bunkers, and soon chokes the condenser tubes as well as the pumps themselves, unless the rose boxes have sufficiently small holes to prevent it; and when the rose-box holes are so small as to prevent small coal passing through them, they soon choke, and render the pumps useless for the time. With a centrifugal circulating pump there is no danger of the pump choking or getting out of order, and by having a direct discharge overboard, the condenser cannot get choked.

Water Service.—An arrangement of water service is provided in every ship, so that water can be applied to all bearings and slides, and also that a hose may be used in case of fire or any other emergency. This generally consists of a main pipe from a sea-cock leading into a convenient position, and having branches with a stand-pipe and brackets at each main bearing, crank-pin, and group of eccentrics. The water for the tunnel-bearings and thrust-bearing is usually taken from the inside of the stern-tube, thereby serving the additional purpose of circulation in the tube.

Expansion Joints.—Great care should be taken that all pipes conveying hot water or steam, and thereby liable to great change of temperature, should have provision made for the consequent expansion. Expansion joints of all kinds are, however, objectionable on one or two grounds, and therefore should be avoided unless absolutely necessary. If small pipes have bends between the rigid connections, the expansion is allowed for by the variation in their curvature. If the rigid connections are far away, care should be taken to prevent the pipe from assuming full curvature to the circle, and thereby tear away from the flanges; but in all large pipes, and wherever the bends are but very slight, expansion joints should be fitted, for very serious accidents have occurred in consequence of their absence. When the length of piping to undergo expansion is not great, the ordinary bellows joint is preferable, as being free from any liability to leak; but when the expansion is very great an ordinary stuffing-box and gland, or Fawcett joint, must be resorted to.* This latter should be avoided for exhaust pipes to a condenser, as they are very liable in this case to leakage which is not easy to detect. A thick india-rubber jointing ring will generally serve this purpose with eduction pipes, which in the case of large ones may be 2 inches thick.

Again, in ships of light construction, which undergo a considerable amount of racking in a sea-way, no length of pipes should be without some provision for expansion and contraction, and all extra rigidity should be avoided, that they meet the changes in a ship. Great care should be taken to provide means for resisting the thrust of pipes which are subject to internal pressure or heat, otherwise the joints are severely strained, and the connection of pipe to flange and of valve-box to boiler in danger of destruction. This is the more necessary when bends are expected to do duty for expansion-joints, etc. Loose pipes should be always carefully "anchored."

Safety Collars.—It has sometimes happened that pipes have blown out of a Fawcett joint, owing to there being a bend near it on which the steam pressure acted; in all cases of this kind a collar should be brazed to the pipe, not less than 18 inches from the bend, through which bolts pass connecting it to the flange of the stuffing-box on the other pipe.

Flanges.—Few things look worse in an engine-room than heavy broad flanges to the copper pipes, especially when they are quite unnecessary. The breadth of the flange need not exceed three times the diameter of the bolts through it, and its thickness should be proportional to the pitch of the bolts. The pitch of the bolts should be from four to six times their diameter, depending on the steam pressure and thickness of flange.

Table xlviii. has been drawn up as generally indicating the best thickness of pipes for various pressures and purposes.

In the merchant service the main steam pipes, when of copper, are usually from No. 1 to 4 B.W.G. thick, depending on their diameter and the pressure to which they are exposed.

Both in the Navy and often in the mercantile marine all steam pipes over 2 inches diameter are of steel; in the former it is solid cold-drawn, in the latter hot-drawn, or welded with riveted cover-straps for large sizes.

Board of Trade Rules for Pipes.

117. The working pressure of well-made copper pipes when the joints are brazed is found by the following formula:—

$$\frac{6{,}000 \times (T - \frac{1}{16})}{D} = \text{working pressure};$$

T = thickness in inches;
D = inside diameter in inches.

* These are not always satisfactory, for if the packing is tight enough to prevent leakage of high-pressure steam it grips the pipe and prevents sliding. It is found now that a spigot end carefully fitted into a smooth bored socket and made so thin that pressure tightens it is much more satisfactory. (*Vide* fig. 182*a*, p. 510.)

When the pipes are solid drawn and not over 10 inches diameter, substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

118. The internal pressure on steel pipes made of good material, which are lap-welded and are a sound job, may be dete mined by the following formula, provided that the thickness is not less than $\frac{1}{4}$-inch :—

$$T = \text{thickness in inches;}$$
$$D = \text{diameter inside in inches;}$$
$$\frac{6{,}000 \times T}{D} = \text{working pressure.}$$

TABLE XLVIII.—Thickness of Copper Pipes (L.S.G.).

Diameter of Pipe in Inches.	Steam Pipes. Boiler Pressures in Lbs. 200	180	155	125	85	50	Auxiliary Exhaust Pipes.	Waste Steam Pipes.	Main Water Pipes.	Bilge Suction and Disch. Feed Suction and Fire Service.	Diameter of Pipe in Inches.	Main Eduction and Air-Pump Suction.
22	...	...	...	...	0	7	...	...	4	...	35	3
21	...	...	...	...	1	7	...	...	4	...	33	3
20	...	...	...	...	1	7	...	...	5	...	32	4
19	...	...	...	...	2	8	...	...	5	...	29	4
18	...	...	...	4/0	2	8	...	13	5	...	28	5
17	...	...	6/0	3/0	3	8	...	13	6	...	25	5
16	...	7/0	5/0	2/0	3	8	...	14	6	...	24	6
15	...	6/0	4/0	0	4	8	...	14	7	...	21	6
14	7/0	5/0	3/0	0	4	9	11	14	7	...	20	7
13	6/0	4/0	2/0	1	5	9	11	14	7	...	17	7
12	5/0	3/0	0	2	5	9	11	14	8	...	16	8
11	4/0	2/0	1	3	6	9	11	15	8	...	15	8
10	3/0	0	1	3	6	10	11	15	9	...	14	9
9	2/0	1	2	4	7	10	11	15	9	...	13	9
8	1	2	3	5	8	10	12	15	9	...	12	10
7	2	3	4	6	8	11	12	15	10	9	11	10
6	3	4	5	7	9	11	12	15	10	10	10	11
5	5	6	6	8	9	11	12	15	11	10	9	11
4	6	7	7	9	10	12	12	15	11	11	8	12
3	8	9	9	10	11	12	12	15	12	11	7	12
2	10	11	11	11	12	12	12	15	12	12	6	13
1	13	13	13	13	13	13	13	15	13	13	5	13

Blow-off and Scum Pipes.									
Diameter of pipe in inches, .	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3
Thickness, L.S.G., . .	10	10	10	9	9	8	8	7	7

Feed discharge pipes to be as steam pipes for 30 per cent. higher pressure; but in no case to be taken lower than 125 lbs.

Receiver pipes to be as steam pipes for half the test pressure of the cylinder to which they lead steam; but in no case to be taken lower than 50 lbs.

The above gauges refer to straight pipes only; bends to be suitably strengthened.

119. Feed pipes should be made sufficient for a pressure 20 per cent. in excess of the boiler pressure.

The Bureau Veritas Rules for steam pipes are—

(1) If solid drawn copper, $t = \frac{D \times P}{6{,}400} + 0{\cdot}05$ inch.

(2) Steel pipes $= \frac{D \times P}{6{,}850}$; but no pipe to be less than $\frac{7}{32}$-inch thick.

Lloyds Register Rule for wrought-iron and steel pipes is—

$$\text{Working pressure} = C\frac{(t - 0{\cdot}125)}{D}.$$

t is the thickness, and D the internal diameter in inches.
C = 9,000 for lap-welded and 12,000 for seamless pipes.

They are to be tested to three times the working pressure.

The Admiralty specifications allow of steel pipes being used very freely for steam pipes, being solid drawn up to 15 inches, and the larger ones solid drawn or welded and fitted with a riveted cover strap on the weld outside. The flanges are generally riveted on to the large pipes.

In the merchant service cast iron was sometimes used for large pipes conveying water and exhaust steam, and occasionally for the main steam and feed pipes, and where cost and stiffness are of more consideration than weight, they may be used with advantage, for although there seems an element of danger in them, experience proved them to be reliable and safe with even comparatively high pressures of steam (100 lbs.). With the higher steam pressures, however, this material has been dropped for steam and feed pipes, and copper, solid drawn steel, or welded wrought iron substituted. The flanges are sometimes riveted and sometimes brazed or screwed. Some pipes are made with flanges in one piece by an ingenious system of press.

The joints of all pipes should be in such positions as to be easy of access.

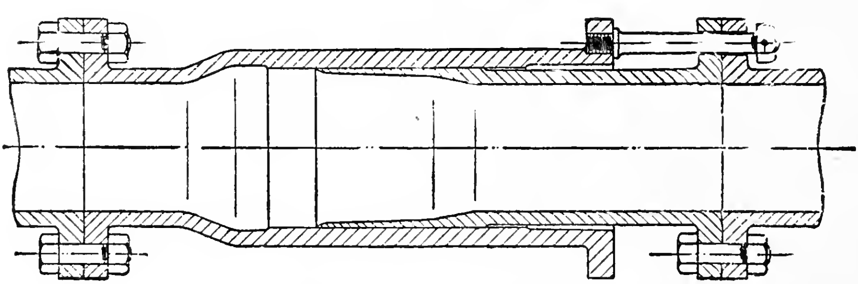


Fig. 182*a*.—Steam and Feed Pipe Expansion Joint.

CHAPTER XX.

AUXILIARY MACHINERY.

In the Early Days of Marine Engineering there was practically no auxiliary machinery whatever, and until quite modern times it has been restricted to such moderate limitations as to require little or no comment, and the demand from such as there was for steam to work it so small as to need no addition to the boiler capacity. Thirty-five years ago the merchant ship had a donkey pump which served almost every purpose, and it was only in large ships that a second general purposes pump was found, besides the ballast pump in ships having water ballast. On deck there were the steam winches, a steam steering gear, and generally a steam windlass in all but quite small ships. In passenger ships there was an electric light engine and sometimes ventilating fans. To-day the auxiliary machinery is both numerous and powerful, so that the demand for steam to supply them all requires considerable additions to the boiler installations, and the consumption of fuel in such ships is an imperfect indication of the economic quality of the main engines.

Auxiliary Machinery on Shipboard to-day requires a knowledge to deal with it and a capacity to take care of it quite special, and its supervision is as important as, and rather more exacting than, that of the main engines. The marine engineer in charge of it must have a technical knowledge sufficiently good and general to enable him to overhaul, adjust, or repair all such mechanisms and fittings, but it is not necessary that he should be able to design and construct them, that is the métier of the specialists engaged in their manufacture.

There are Three Distinct Classes of Auxiliary Machinery:—

(1) Those things which are absolutely necessary to and are practically part of the main engines, such as—

The centrifugal circulating pump and its engine.
The special air-pumps and engines or motors when separate from the main engine.
The feed-pumps and auxiliary or supplementary feed-pumps.
The steam reversing and stop valve gears.

(2) Those machines which are incidental to the machinery whereby the working is rendered more efficient and economical, such as—

The steam-turning gear, and any winches or hoists in the engine room.
Ventilating fans and forced-draught fans.
Steam ash hoists.
Electric light and power engines and dynamos.
Steam gear for operating fire-bars, etc.

(3) Those engines and machines which are incidental to and requisite for the ship, such as—

The steam steering-gear.
Steam winches.
Steam or hydraulic cranes.
Steam windlass and capstans.
Baggage and boat hoists.
Refrigerating machinery and plant.
Hydraulic engine and pumps for hydraulic cranes and hoists.
Bilge pumps and ballast pumps.
Ventilating fans.
Sanitary pumps.

(*a*) **On Warships there are** in addition to most of the preceding—

Air compressing machines for torpedoes, etc.
Ammunition hoists. Coaling machinery.
Gun training machinery. Boat hoisting winches.

(*b*) **Where Oil Fuel is used** there must be, in addition—

Pumps for dealing with the storing of the fuel.
Pumps for supplying it to the boilers.

In addition to all these things there are other fittings which may come under the category of auxiliary machinery, as—

Feed filters.
Feed heaters. Lubricating oil pumps.
Feed distillers and fresh drinking water distillers.
Auxiliary and winch condensers. Auxiliary boilers.

The consumption of steam by the auxiliaries in a warship when under working conditions is enormous, for in a ship such as the second-class cruiser "Diana," with ordinary cylindrical boilers, in 1899, it was at the rate of 4,544 lbs. per hour when cruising at slow speed, while at high speed ($\frac{4}{5}$ full power) it was as much as 16,384 lbs. per hour, or 12·5 per cent. of all the steam generated, and at full power it was 10 per cent. The waste in this ship necessitated the distillation of 1,792 lbs. of water per hour to make it up. For the ship's use for domestic purposes 560 lbs. per hour were required, so that the distillation of 2,352 lbs. of water per hour has to go on regularly to keep such a ship going when working at high speed.

The consumption of steam for auxiliary machinery on a modern express turbine-driven steamer is enormous, and when on service with a full complement of passengers the amount required for heating and distilling fresh water for washing and domestic purposes is very large. Mr. Bell has shown that on the "Lusitania" during her trials, and, therefore, practically without passengers, when developing her full sea power of 65,000 S.H.P., there were pumped to the boilers no less than 998,000 lbs. of water per hour, and he found as follows :—

The main turbine engines used	851,500	lbs. of steam,	or 13·1	lbs. per S.H.P. hour.
The auxiliary machinery,	114,000	,,	= 1·75	,,
For evaporating in distilleries,	32,500	,,	= 0·50	,,
Total consumption,	998,000	,,	= 15·35	,,

It will be seen from this that of the steam generated by the boilers, the auxiliaries used no less than 11·42 per cent., and the main engines propelling the ship 85·32 per cent. The following Table xlix. shows the consumption of steam fuel, etc., on the "Lusitania" on service conditions at different speeds, and is interesting as illustrating one of the economic arguments in favour of high speed for the Atlantic service. Mr. Morison reckons that the demands of the auxiliaries in the engine department (including steering gear) of an ordinary merchant ship amount to 7½ per cent. of the total steam production, and, therefore, demanding more attention than has been hitherto bestowed on them in that class of ship.

TABLE XLIX.—Steam and Fuel Consumptions of R.M.S. "Lusitania" running on Sea-service Conditions at Various Speeds.

Speed of ship in knots per hour,	15·77	18·00	21·00	23·00	25·40
Shaft horse-power, ...	13,400	20,500	33,000	48,000	68,850
Consumption of steam per hour, total, lbs.	284,500	253,600	493,300	668,300	879,500
" " " auxiliaries, lbs.	71,000	76,400	85,700	96,700	116,500
Consumption of auxiliaries per cent. of total, ...	25·0	21·6	17·4	14·51	13·2
Steam consumption per S.H.P. hour, turbines, ... lbs.	21·23	17·24	14·91	13·92	12·77
Steam consumption per S.H.P. hour, auxiliaries, ... lbs.	5·30	3·72	2·60	2·01	1·69
Steam consumption per S.H.P. hour, total, ... lbs.	26·53	20·96	17·51	15·93	14·46
Coal consumed per S.H.P. per hour, total, ... lbs.	2·52	2·01	1·68	1·56	1·43
Total consumption of fuel on voyage, 3,100 miles, . . . tons	2,980	3,190	3,670	4,520	5,390

It will be seen by the above that, taking auxiliaries into account, as well as the increase in consumption of the main engines at slow speeds, a horse-power at 25·4 knots costs only 57 per cent. of that at 15·77 knots and 71 per cent. of that at 18 knots. If domestic demands be taken account of the difference is still greater, so that the S.H.P. then is considerably less than 57 per cent. in cost.

In the ordinary passenger steamer the demand for fresh water for engine requirements and domestic purposes is always great, and in ships trading in cold climates steam heating is resorted to, which is another drag on the boilers.

It will be understood, then, that in the modern passenger steamer and warship the boilers must be considerably in excess of the requirements of the main engines, even to the extent of 10 to 15 per cent., and in small express ships probably greater.

The Exhaust Steam from all auxiliaries should be retained in the system, and be delivered to a feed heater or else to a special condenser, where it is converted into feed-water for the boilers. When the exhaust is delivered to the main condenser, it is better that it shall not be done direct, so that all auxiliaries may then be worked with a back pressure above instead of below that of the atmosphere, for in the latter case the vacuum may be seriously affected by the air leaks from winches and such like things as are not immediately under the engineering staff's notice, and liable to be in want of adjustment of glands; it is better, therefore, that these work with a back pressure above atmospheric pressure, and their exhaust enter the L.P. cylinder valve box of the main engine. or the L.P. turbine if so fitted.

Mr. D. Morison has shown that in an ordinary cargo steamer conveying passengers, and having engines of 5,000 I.H.P., whereas the main engines alone consumed 70,000 lbs. of steam per hour, the auxiliaries consumed 8,650 lbs., or 12·36 per cent. of the main engine demand, and 11·0 per cent. of the total supply to machinery, and was distributed among them roughly as follows:—

Consumed by the	evaporators to make up waste,	2,800 lbs.
,,	steam-steering engine,	1,000 ,,
,,	feed-pumps,	950 ,,
,,	centrifugal circulating pumps,	1,800 ,,
,,	fan engines on boilers, etc.,	1,300 ,,
,,	electric light engine,	800 ,,

In an ordinary cargo steamer of 1,500 I.H.P. and a consumption in the main engines of 21,000 lbs., the auxiliaries require 1,660 lbs. per hour, or 7·9 per cent. of that of the main engines, and 7·32 per cent. of the total generated.

Under these circumstances it is quite requisite that means be adopted whereby the demand for these necessary parasites be most carefully considered and reduced to as small an amount as possible; but what is perhaps the better policy, since the supply must be always large, is to make the best use of the heat rejected by these engines. Mr. Morison points out that by exhausting to the condenser 80 per cent. of the available heat is lost, and even if the exhaust steam from them is passed through the L.P. cylinder of the main engines the loss is almost as great, whereas if it is used to heat the feed-water there is no practical loss, and even with a very high vacuum in the main condenser, and the consequent low temperature of the hot well, the feed-water, as delivered to the boilers, will be quite hot.

It is estimated that from such sources in the 5,000 I.H.P. ship as much as $11\frac{3}{4}$ millions of thermal units are available for the purpose, and that thereby the water as coming from the condenser having a vacuum of 28·25 inches, with a temperature of 87° F., only will be raised to 211° F. after mixing and before delivery to the boiler. Further, that in the cargo steamer of 1,500 I.H.P. the available heat is nearly $2\frac{1}{2}$ millions B.T.U., and the feed-water will be raised from 87° to 175° F. by mixing before delivery to the boilers. But seeing that the ordinary feed-pumps on shipboard can draw water only when the temperature is below 170°, special means must, of course, be provided for dealing with water so much warmer. This, however, is quite a simple matter, and inexpensive considering the great gain to be made thereby.

Some of the Auxiliaries work constantly at a fairly uniform rate, and when the main engines are working at full speed these auxiliaries also proceed at nearly full power; consequently they may have a compound arrangement of cylinders or other means of using steam at a fairly high rate of expansion.

and provision for overload when occasion demands more than their normal full output permits of their cylinders being comparatively small. The circulating pump, for example, might be always driven by such an engine; likewise the air-pump, feed-pump, etc., draught and ventilating fans, and such like machines.

Other Auxiliaries work intermittently and at full speed only for short periods; they may require then to give a big output compared with that under normal conditions. The steam-steering gear, for example, may stand for hours with hardly a movement, and during the day on the high seas in fine weather do very little work altogether, and at no one time develop more than 10 per cent. of its maximum power. For such work the small cylinder with the late cut-off answers quite well. For the windlass, capstans, etc., it is not worth while, for the short time they work, to have any complications for the sake of economy. The winches and cranes are abominably extravagant with steam, and seeing the number of hours they work in many cargo steamers, it is worth while studying means for a reduction. Attempts have been made from time to time by supplying them with compound cylinders; but the wear and tear on them is so great, and in the past winches are not heeded very much so long as they will somehow do the work required by the stevedore, that small encouragement has been given to these elementary efforts. Still, it is quite worth the while of the shipowner to have a much better sort of steam or electric winch, to work with less than half the steam and half the wear and tear; but such an instrument cannot be turned out at the exceedingly low price of the present winches.

The Electric Light Engines also are important enough to have bestowed on them more consideration than has obtained often in the past. They, too, have intermittent work, but they are running every day that the ship is on service, and during the early hours of the night the demand on them in passenger ships is very heavy. There is every reason for their having compound cylinders, that a fair economic rate of expansion may be obtained from steam of the pressure of the boilers; in fact, in large ships some few of these engines might be triple compound with advantage, or else what is much the same thing, two-stage compound with a L.P. turbine connected to the exhaust. In most ships electric engines must be run all day for one purpose or another, inasmuch as so far the storage batteries are not a success; they are heavy, costly, liable to get out of order, and the acid fumes from them a nuisance. It is also imperative that in passenger ships there must be provision for such a contingency as a break-down or stoppage; nowadays the enclosed forced lubricated engine (figs. 188, 189) can be relied on to keep going for an indefinite period without stopping, and the wear and tear on them is a negligible amount; but the dynamo is still sensitive to water and water vapour. There should be, therefore, a stand-by unit in every ship,* or the full requirements divided into two or more units, rather than have one engine and dynamo only. In large passenger steamers the full requirements at night time necessitate the running of all but one or two units, which are held in reserve; as the lights are put out the other units are one after the other stopped, until the load is easily carried by those left running. In a comparatively small steamer there should be two engines and two dynamos, each pair separate and capable of generating 60 to 70 per cent. of the maximum demand, so that the two are run together only for a short

* By preference an oil engine placed well above water to be useful in case the boiler-rooms are flooded.

time, and in case of a break-down with one the other can practically keep the ship well lighted. For a larger ship there should be three units, each capable of doing 55 to 60 per cent. of the maximum, so that only two are required at one time, the third being quite in reserve, and one only during the day, the early and late night. The largest should have four units, each capable of generating 40 per cent., and only three running at full demand. Some very large and important ships have a still greater number of units, and keep two in reserve. It may seem extravagant to have such a large subdivision, but the thermal efficiency of these small engines at full load is much higher than that of larger ones at loads less than the maximum. The mechanical losses also are less in the smaller engines at the same revolutions. In fact, it has to be borne in mind that electric light engines run at constant rate of revolution whatever the load may be, and so instead of an engine whose maximum power is 400 at 450 revolutions per minute doing, say, 100 B.H.P., there is one of only 200 H.P. doing the 100 B.H.P. with less expenditure of fuel and oil, and as it may be running all day, the difference in expenditure is very appreciable. It would be better still to have, say, three units, each of 133 H.P., to do the 100 B.H.P. output, for a margin of 33 per cent. would be ample, and as the fall might be at times to 50 I.H.P., of which 20 only would be B.H.P. or useful power, the cost would be very great compared with that of the 133 B.H.P. engine, the B.H.P. of which for 50 I.H.P. would be probably not less than 39.

The current on shipboard is much lower than is now common for house lighting on shore, being generally only 100 volts at the switchboard, whereas in London 210 is common.

The Working of all Auxiliaries by Electric Current is, no doubt, the ideal thing, for thereby much that is highly objectionable in the present ordinary arrangement is avoided, and all that is desirable and worth having is accomplished. The power is then generated by a highly efficient and economic engine placed under the immediate observation and control of the skilled and responsible engineers on watch; the energy is transmitted by means of cables, which may be led throughout the ship in the easiest way possible, and when lead-covered and otherwise protected from harm are both safe and efficient transmitters. The motors are simple, light, and easily fixed machines, which can be placed in any position in quite a contracted space, and protected, water-tight, etc. Their efficiency is much higher than that of any of the steam engines employed for similar purposes on shipboard, and they require little or no attention from skilled or other attendants. Winches, capstans, windlasses, steering gears, pumps, etc., can all be worked by motors, and avoid the transmission of steam more or less wet to distant parts of the ship through pipes which, even when well lagged, leak an appreciable amount of heat, and are always a nuisance to designers, ship officers, and all having to do with them. Their exhausts are a worse nuisance still, for if overboard the ship's deck is often smothered in steam, if to the main condenser the vacuum is often affected adversely, and the only thing is to increase the back pressure, and thereby reduce their efficiency, or conduct their exhaust steam to a separate condenser with much loss of energy thereby. There is another objection to steam for deck auxiliaries in winter time—that is, that the small amount of leakage past the stop valves permits of some condensation in the deck pipes, and the water so formed may in severe cold weather freeze and prevent the working of the gear at an inopportune time.

On the other hand, an electric installation is a somewhat expensive one, and it is urged by some that it is dangerous from cut-outs fusing and from short-circuiting due to water. This is all quite true as a liability of the system, but if proper care is taken in the selection of materials, and in getting the installation carried out by experienced men, the risk of any serious damage arising from it is now only small.

There is One Cause of Serious Loss of Steam which is often overlooked, and that is the use of it in whistles and syrens to give notice to other steamers, etc. In such a ship as the "Lusitania" the whistle consumption as a percentage of the total steam generated is, of course, very small, but considering

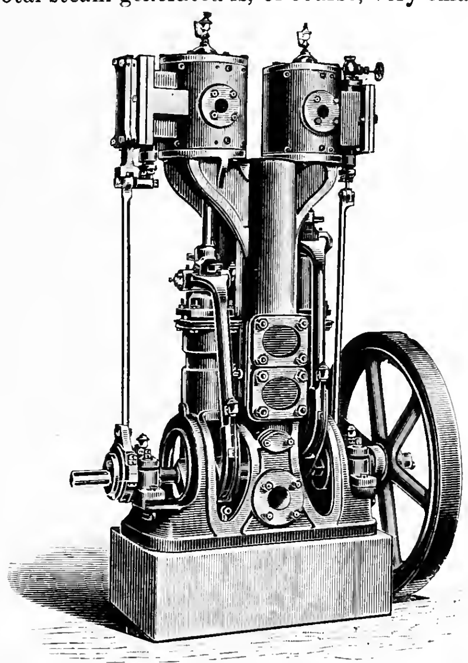


Fig. 183.—Double Ram Flywheel Feed-Pump.

a tramp steamer with a generating power less than 2 per cent. of the "Lusitania" has to make the same noise, and probably oftener, during the twenty-four hours, the loss to her must be considerable and serious. It would seem, therefore, that compressed air either in bottles or stored in a receiver from a pump driven by the main engines would answer the purpose even better than wet steam, and cost much less. Compare the expenditure of air by a bugle in producing a noise which can be heard as far and as clearly as many ships' whistles with the steam required for such of them as evidenced by the cloud flowing away at each blast. It must be remembered also that all day long steam is being condensed in the whistle pipe, and if the resultant water is not blown out, to the damage of the funnel paint and deck fittings, it runs back minus its latent heat. Steam whistles, therefore, waste much heat and much water, both of which cost the shipowner money.

The Feed-Pumps, both main and auxiliary, are now in naval and express steamships worked independently of the main engines, but in cargo steamers the feed- and bilge-pumps are still worked f om the air-pump crosshead of

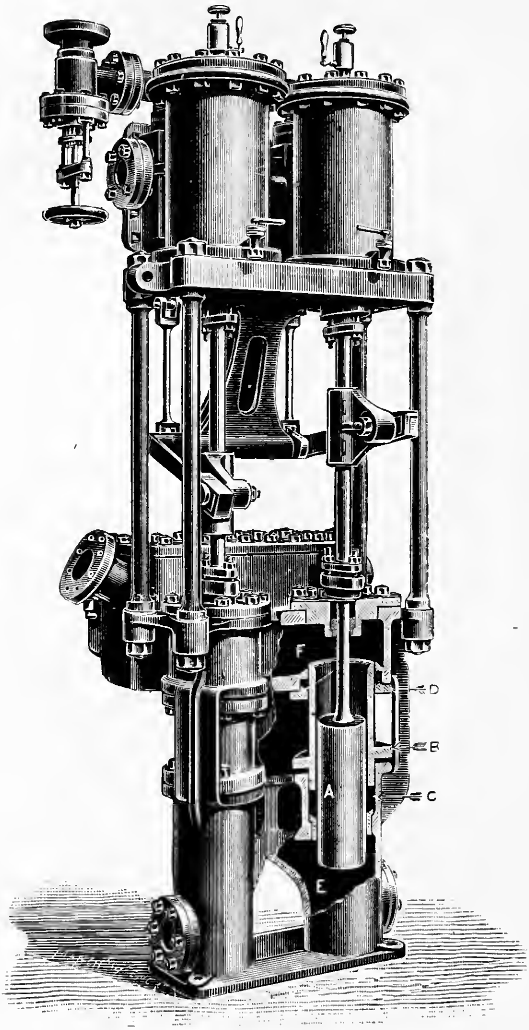


Fig. 184.—External Packed Pumps (Duplex).

reciprocating engines. There are two kinds of such pumps, the flywheel type, in which the pump is driven direct from the steam cylinder piston, but controlled by a flywheel, etc., as seen in fig 183, and the direct-driven without flywheel, as shown in figs. 184 and 185. The flywheel pump is not so much fancied now as formerly; it is, however, a good working one, and always makes its full stroke; the pump rams are packed externally, and, therefore, kept in a good state of tightness; the cut-off in the steam cylinder

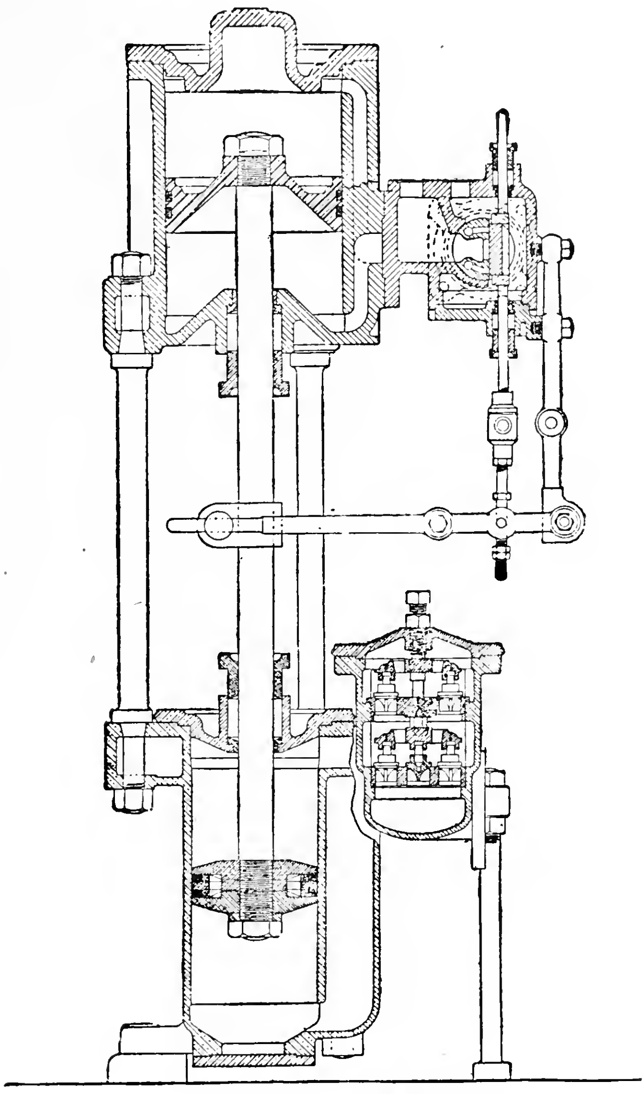


Fig. 185.—Section of a Weir Pump.

may be comparatively early, whereas with the others it is very late, and the clearance very great compared with the flywheel one. On the other hand, such pumps as shown can be worked "dead slow" or at high speed with equal ease. and be quite automatic and under control. For quite small ships, where there is only one, or at most two, auxiliary pumps for general purposes, the duplex flywheel pump (fig. 183) is a very good and

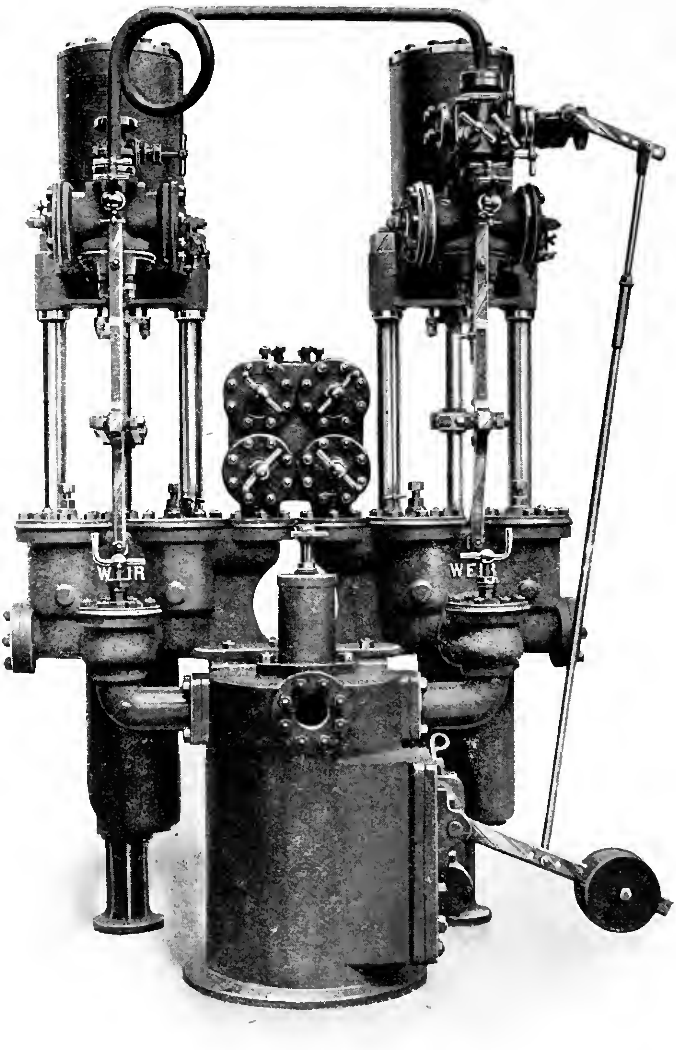


Fig. 186.—Weir's D. A. Feed Pumps, with Float Tank and Automatic Gear.

reliable one. For larger sizes, and especially where the feeding is all done by independent pumps, the type invented and successfully developed by Mr. James Weir (fig. 185) is in general use. Fig. 186 shows a pair of such pumps fitted with a "float" chamber and gear, so that their operation is under complete control. By this arrangement the pumps are slowed down on the supply of water from the hot well failing and stopping altogether before the supply is so low as to risk the indraught of air. So long as the water is supplied to them from the hot well or reserve tanks they will continue to deliver to the boilers, and should the water tenders in the stokeholes check the supply, the pumps will slow down under the increased resistance.

Fig. 184 is the vertical variety of the well-known and popular duplex pump introduced by the Worthington Company many years ago, and improved in this country, as shown by the arrangement whereby the plungers are packed externally. Such pumps as these have the merit of cheapness, and when pressures are not high the ordinary piston (pump) type work very well, but with tightly packed pistons or plungers there is a tendency to fail making the full stroke every time. The Weir, like the Worthington types of pumps, are somewhat wasteful of steam, but have high mechanical efficiency as a set off; moreover, they require little or no oil and few repairs, so that altogether the Weir pump, even when without compound cylinders, is a favourite with sea-going engineers who appreciate their reliability to continue doing steady duty with no attention, when the demand on their time is for other and more important things. In fact, all auxiliaries of this kind should be automatic and reliable, and require no watching at all, but only the casual attention from the greasers on their rounds.

Engine-room Pumps are nowadays very numerous, as may be seen on the plan of R.M.S. "Olympic." There are, besides these feed and auxiliary feeds, other similar pumps to deliver water on every deck in case of fire, and for daily use in washing them; pumps to supply closet and other domestic tanks, baths, boilers, etc. Pumps to keep the bilges of the engine-room free from dirty water, oil, etc.; to draw from the bilges of every hold and compartment; to draw from reserve fresh-water tanks, and from ballast and trimming tanks; and most of these many pumps must necessarily be in duplicate.

Steam Reversing Gears are dealt with elsewhere, and those gears now used for operating the massive stop and pass valves of a turbine installation are in all essentials like the Brown reversing pull-and-push engine (fig. 266).

Ventilating and Forced-draught Fans are now in very general use, and are often driven by electric motors in preference to a steam engine, as they are often in out of the way, inaccessible places, and require stopping and starting by unskilled attendants. (For Table, see p. 522.)

Electric Light and Power Engines are now fitted to almost all ships, inasmuch as the light obtained thereby is better than that of any oil lamps, it is cheaper too, and when the installation is carried out in a proper way it is much safer and cleaner. In the passenger steamer and warship electric power transmission, as well as lighting, is of great advantage. It is, however, highly desirable that such engines as are employed for this purpose shall be fairly economic in steam, but, above all things, capable of running continuously without variation in speed or trouble of any kind day after day, night after night, without any attention from the staff beyond a casual visit. For this purpose the compound-enclosed self-lubricating engines patented and introduced by Bellis & Morcom have proved most satisfactory, and continue in demand. Fig. 187 shows in section the two-crank compound engine as first brought out by that firm, having an ingenious arrangement of central valve, which serves both cylinders, and answers admirably for engines of moderate size; their improved engine, as shown in figs. 189 and 189*a*, is, however, a better one for larger sizes, and is generally more economic of steam. Fig. 188 is a tandem compound single-crank form of engine occupying little space, and used largely in destroyers and small merchant steamers. It is a good steady worker, and gives much satisfaction to those having charge of them. The Admiralty have tried heavy oil engines for driving dynamos in port when steam is not available; they may be of the Diesel or semi-Diesel type; they are somewhat noisy, and unless care is taken when using certain oils the exhaust smells rather badly. Turbines have also been tried for electric light generating in certain ships, but the high revolutions of such as have been used have proved trying to the dynamos, and, therefore, not altogether satisfactory; moreover, the small turbine is never so economic in steam as the small compound reciprocator, and it is not until a power of at least 1,000 horse that the turbine equals the reciprocator, and further, whereas the latter remains fairly economical when exhausting to the air, the turbine does not. The space occupied by a turbine is so small and the head room so much less than required for a vertical reciprocator that in naval ships, as also in some express steamers, turbines are of advantage. Fig. 190 shows the combination of turbine and dynamo specially designed by Belliss & Morcom for ship lighting purposes.

TABLE L.—PARTICULARS OF ADMIRALTY TYPE OF FANS AND ENGINES FOR FORCED DRAUGHT.

Diameter of Impeller.	Width of Impeller.	Revolutions per Minute.	Cubic Feet of Air per Minute.	Air Pressure W.G.	Effective H.P. required to Drive.	Size of Engine at 180 Lbs. Pressure.		Approximate Weight without Fan Case.
						Diameter of Cylinder.	Stroke.	
Ft. Ins.	Inches.			Inches				Cwts.
2 0	3½	950	4,500	2	3	2½	2½	3¼
2 6	4	800	6,500	2	4	2½	2½	3½
3 0	4½	650	8,800	2	6	3½	3	4¼
3 6	5	550	11,000	2	7	3½	3	4½
4 0	6	500	15,000	2	9	4½	3½	6
4 6	7	450	20,000	2	12	4½	3½	6¼
5 0	8	400	26,000	2	14	5	4	8
5 6	9	350	32,000	2	18	5	4	8¼
6 0	10	320	39,000	2	22	6	5	11¼
6 6	11	300	46,000	2	26	6	5	11¾
7 0	12	275	55,000	2	30	7	6	15½
7 6	15	250	74,000	2	40	7	6	16
8 0	18	250	95,000	2	50	8	7	19½

Some care should be always exercised in placing these high-speed engines that there can be no lodgment of water from condensation, priming, or other

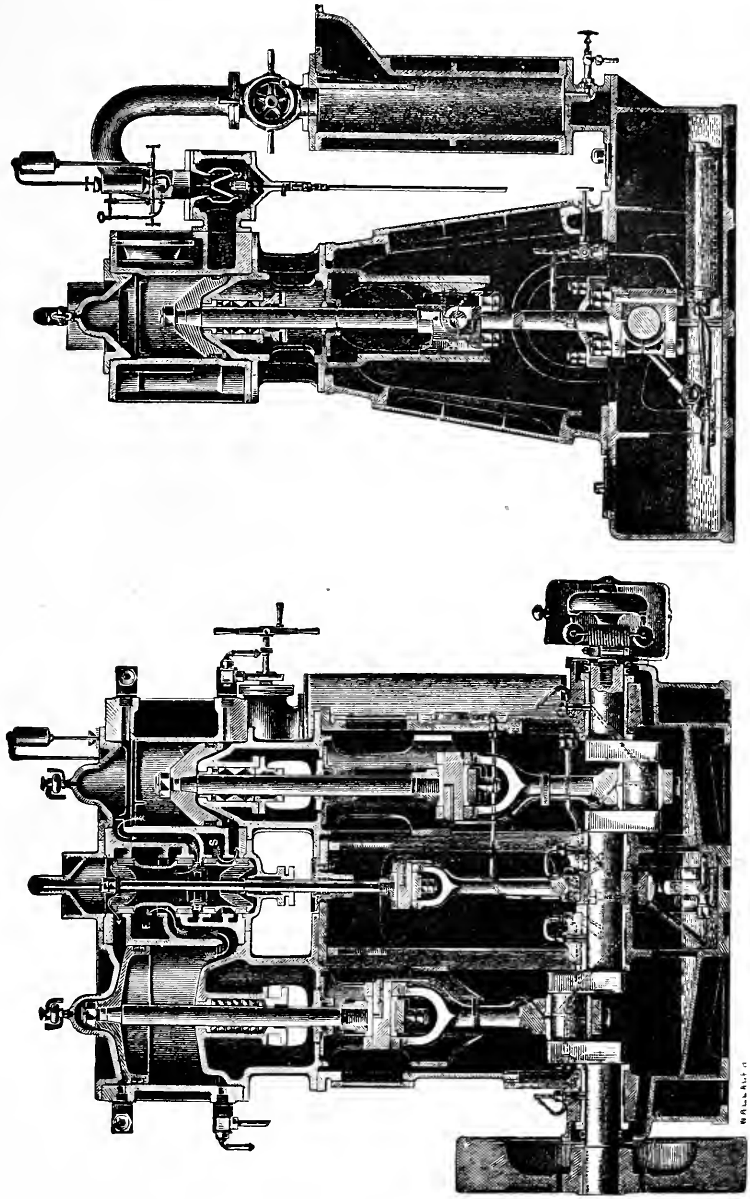


Fig. 187.—Bellis & Morcom's Central Valve Enclosed Compound Engine for Electric Lighting.

cause either previous to or during the working; and equal care taken that no water can enter the L.P. cylinder from the exhaust pipe. In the past it has been only too often the custom to suppose that an auxiliary can be placed

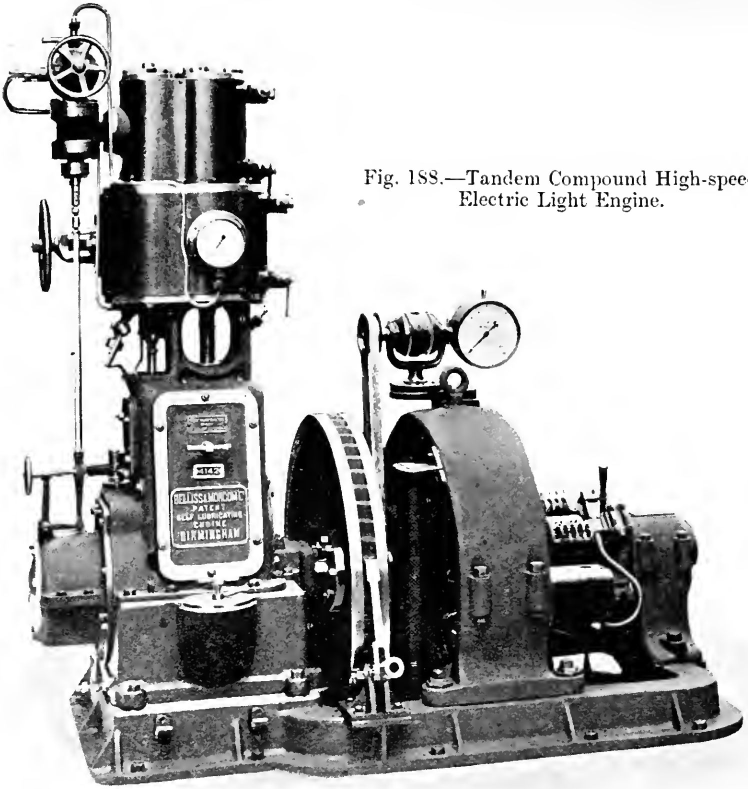


Fig. 188.—Tandem Compound High-speed Electric Light Engine.

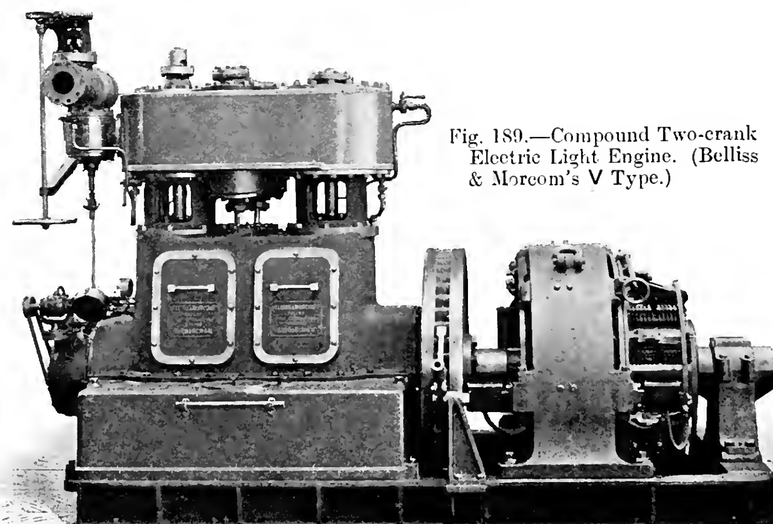


Fig. 189.—Compound Two-crank Electric Light Engine. (Belliss & Morcom's V Type.)

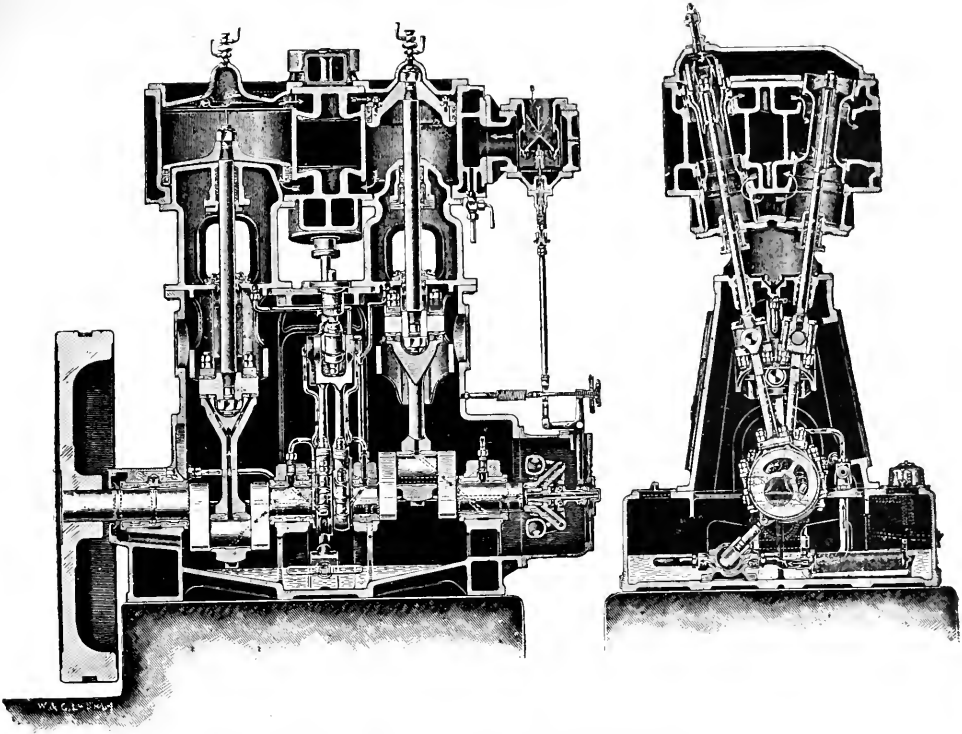


Fig. 189*a*.—Naval Electric Generating Engine.
Self-lubricating Two-crank Compound Inclined Valves. (Belliss & Morcom.)

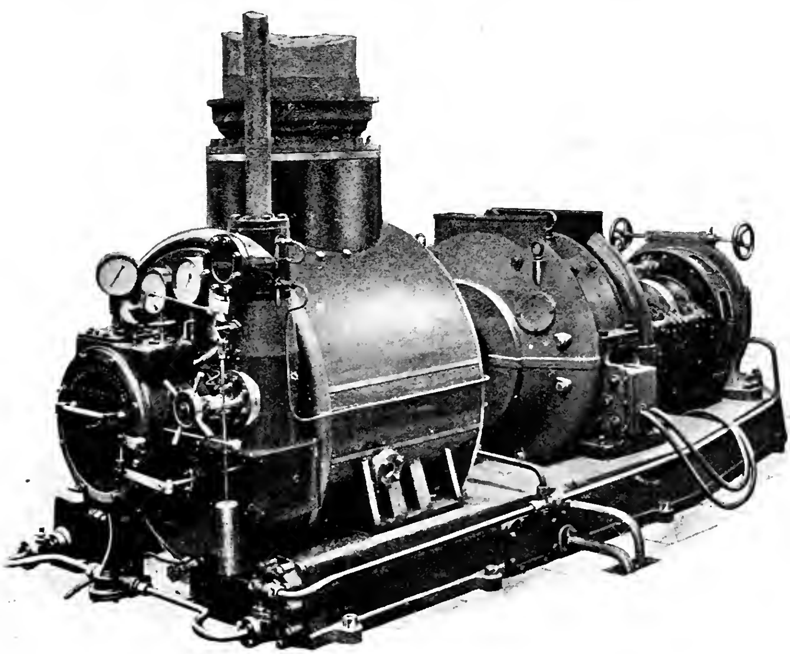


Fig. 190.—Turbine-driven Naval Electric Generating Set, 200/300 kw.

in any convenient corner, and while this may be true of a slow working pump, it is not so of a high-speed engine.

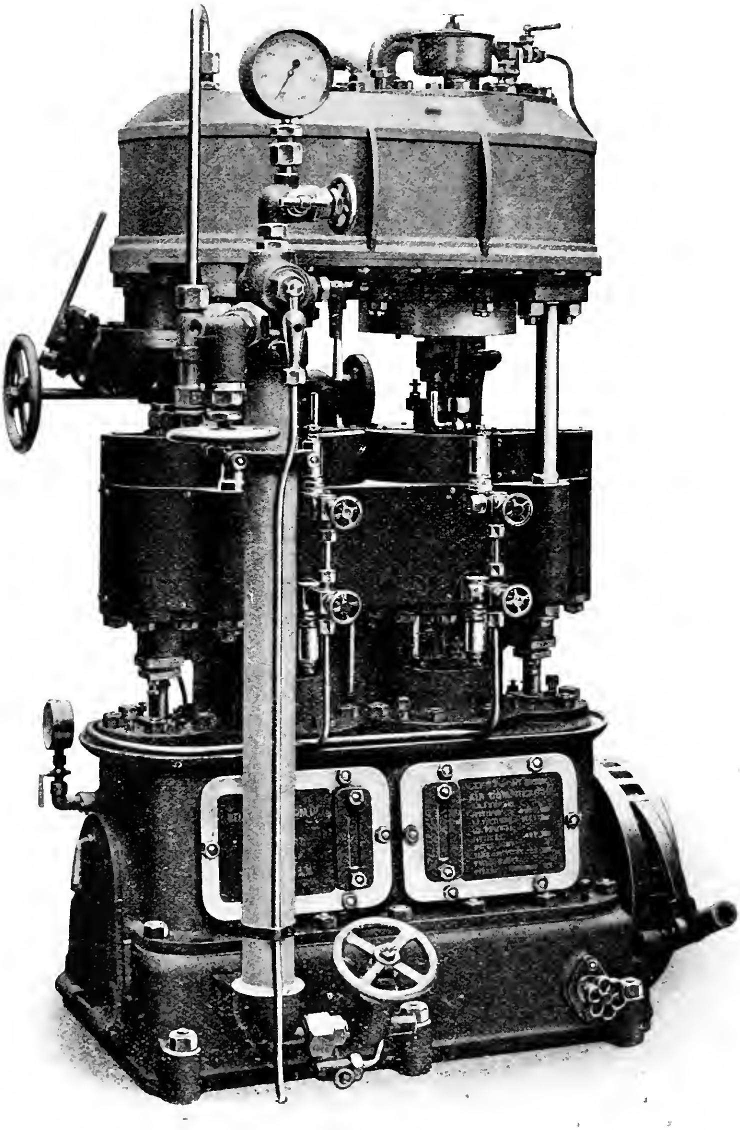


Fig. 191.—Navy Air Compressor (Belliss).

The Hydraulic Pumps and Accumulators are fitted in those ships where energy is transmitted by water to cranes, capstans, and winches, and generally placed in or near the engine-room. In certain trades such means are much better served by water than by steam, and the working is much steadier and safer. In the tropics and sub-tropical regions, where there is no danger of freezing, and also in temperate zones with a mixture of glycerine and water, hydraulic cranes, etc., are most satisfactory; they are desirable as against the noisy, wasteful winch with its heavy wear and tear and comparatively short life.

In the Navy the Air Compressor is a most important machine, inasmuch as by its means the deadly torpedo is endowed with life and speed. Air is drawn from the atmosphere and delivered to the necessary tubes at a pressure of 2,500 lbs. per square inch at a safe and comparatively low temperature.

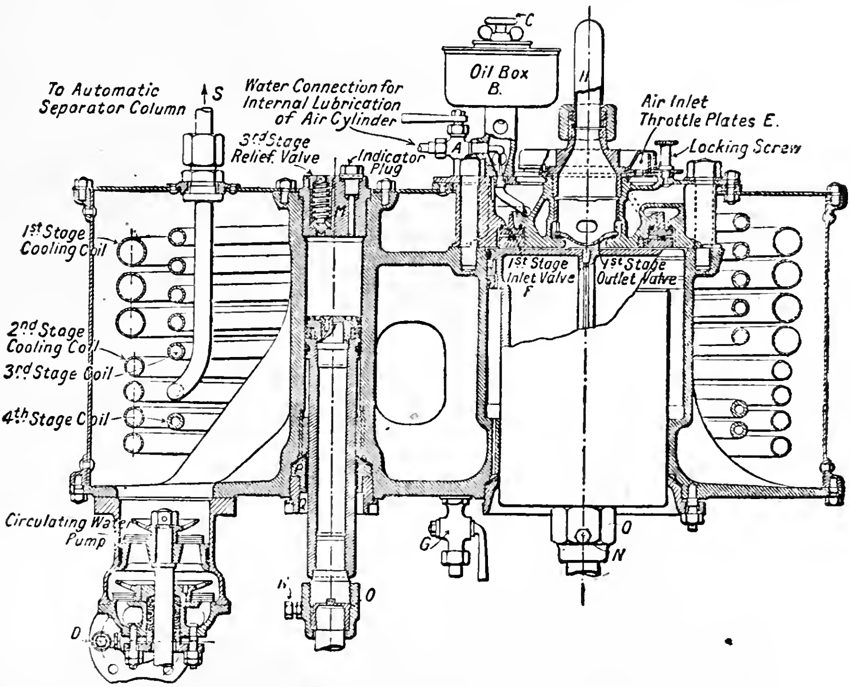


Fig. 191*a*.—Section through Pumps of a Naval High-pressure Air Compressor for 2,500 lbs. per square inch.
(Belliss & Morcom.)

Fig. 191 is the compressor supplied by Belliss & Morcom for battleships and big cruisers worked by a steam engine. In this case the lower portion is simply an enclosed self-lubricating steam engine. Superimposed on it is a compound four-stage system of compressors, whose pistons are connected direct to the engine piston-rods. As the air is compressed it is cooled by coils containing cold sea-water which is passed through them. This machine, when running at 350 revolutions per minute, can deliver 30 cubic feet of air compressed to 2,500 lbs. pressure (*v.* fig. 191*a*). By replacing the steam cylinders with the compressors and connecting an electric motor to the crank-shaft next the flywheel an electric-driven machine is provided, and in this form is sometimes preferred.

Steam Steering Gears are now fitted to quite small ships, and although designed originally by the late J. Macfarlane Gray for the "Great Eastern"

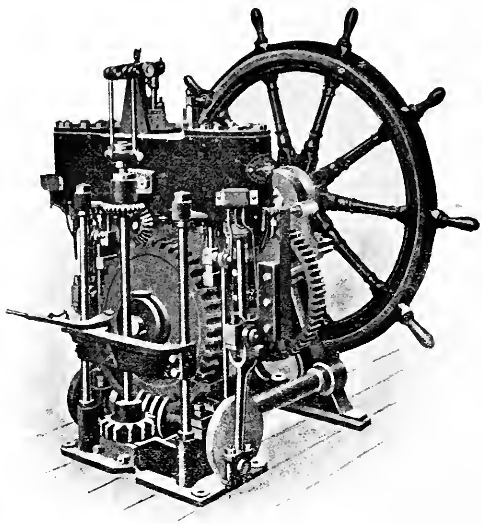


Fig. 192.—Vertical Combined Steam and Hand Steering Gear.
(Bow, M'Lachlan & Co.)

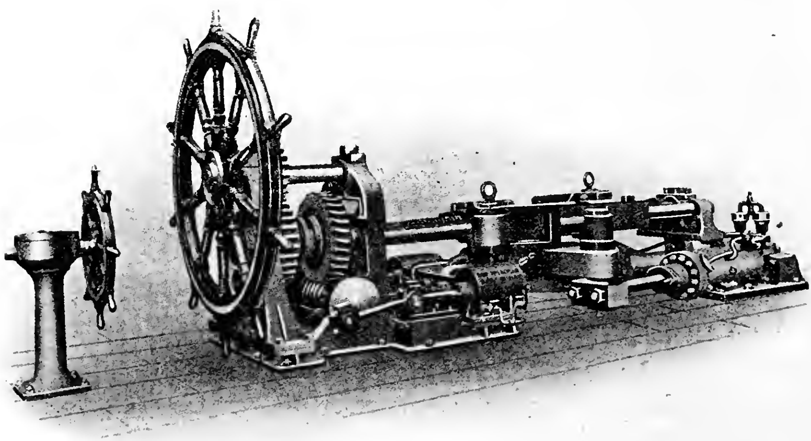


Fig. 193.—Combined Hand and Steam Steering Gear as fitted in High-speed Cross Channel Steamers.
(Bow, M'Lachlan & Co.)

and other very large vessels, which could not be controlled or properly navigated by hand gear, and improved by the late A. B. Brown for all classes of large ships to save the hard work of a large number of men (it would be the whole watch of a modern tramp steamer). Fig. 192 is an example of the kind of steering engine that has been evolved in course of time for use in the ordinary merchant steamer of small and moderate size; it will be observed that the same wheel with which the steam gear is operated when under steam may be employed to steer by hand in case of mishap to it by simply slipping a clutch into gear with the pinion geared into the spur wheel on the chain or wire rope barrel shaft. In a general way in fine weather on the open sea many ship captains prefer hand steering; with such an arrange-

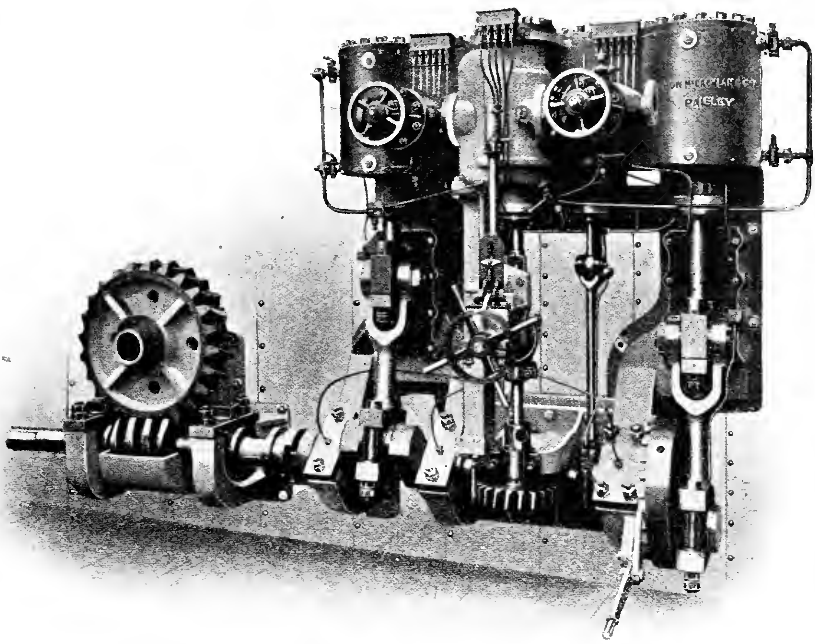


Fig 194.—Type of Vertical Steam Steering Gear fitted in War Vessels.

ment this can be done, and on quite short notice changed to steam gear on an emergency. The compensating gear, whereby the valve displaced by the steersman is brought back to mid or normal position is seen on the front, and deriving its motion from the worm drive. Fig. 193 is a new arrangement as adopted in the modern high-speed turbine steamer, and placed at the stern in direct connection with the well-known right- and left-handed screw arrangement for working by hand. The engine is controlled from the bridge by means of a Brown's telemotor (by preference) or mechanical gearing. Fig. 194 is the vertical type of steering engine as required by the Admiralty for cruisers, and fixed to a convenient bulkhead under water, and protected from damage by shot or shell. In all these cases the speed of the engine is reduced to that of the gear by means of a worm and wheel, as being the most convenient if not the most efficient method. Such worms should work in an oil bath, and run on specially cut wheels with the least possible friction.

Brown's Patent Steam Tiller.—The importance of fitting steamships, whether of high or low speed, with absolutely reliable steering gear cannot be overrated; but in many cases commercial considerations are allowed to weigh in the design of this vital adjunct to a steamer's equipment. By far the greatest number of steamships have their steering engines placed near the bridge, the communication being made with the quadrant aft by means of chains, rods, or wire ropes, with or without spring buffers to take off the shock of a heavy sea. In conjunction with this, hand gear is fitted aft, having double screws with nuts and cross-head, the mode of connection being by pins dropping into connecting links, or by a clutch working on the rudder-head and engaging the cross-head. The trouble involved in keeping these steering ropes or rods properly adjusted and the various pulleys properly oiled, as well as danger arising from the ropes being carried away, has brought about a change in more recent applications of steering gear. The steering engine is placed aft, being coupled by right- and left-hand screws, and in a variety of other ways, direct to the rudder-head; communication from the steering valve being made by a line of shafting to the bridge, thus dispensing with the objectionable rope or rod communication, which is, in the first-mentioned system, subject to the full rudder strains.

An ideally perfect steering gear should fulfil the following conditions:—

1. The steering engine should be attached to the rudder-head without the intervention of chains or ropes.
2. It should let go the rudder when unduly strained, and, when the abnormal strain has gone, return automatically to its former position.
3. The connection from steam to hand gear, and *vice versâ*, should be affected without the use of jaw clutches or the shipping of bolts into holes —which operations are difficult to effect when the ship is rolling at sea with the rudder adrift.
4. The communication from the bridge to the machinery aft should be of a kind which dispenses with rods, chains, and shafting, all being equally troublesome to the shipbuilder to arrange, and to the officers of the ship to keep in order.

With reference to condition 3, it is a common practice to fit rudder brakes on ships where clutches are the means of connection; but as simplicity and fewness of parts are of first importance in steering gear, it is better that such a connection between the steering engine and the rudder, or the hand gear and the rudder, should be one which will act both as a clutch and a brake.

To meet these conditions as far as possible, the steam tiller was designed by the late Andrew B. Brown. Fig. 195 shows an elevation with hand steering gear and plan.

The prominent feature of this gear, in which it differs from all others, is that advantage is taken of as long a lever as will reach from the rudder-head to the limits of the poop deck, which, in the greatest number of ships, varies from 7 to 10 feet, and in the

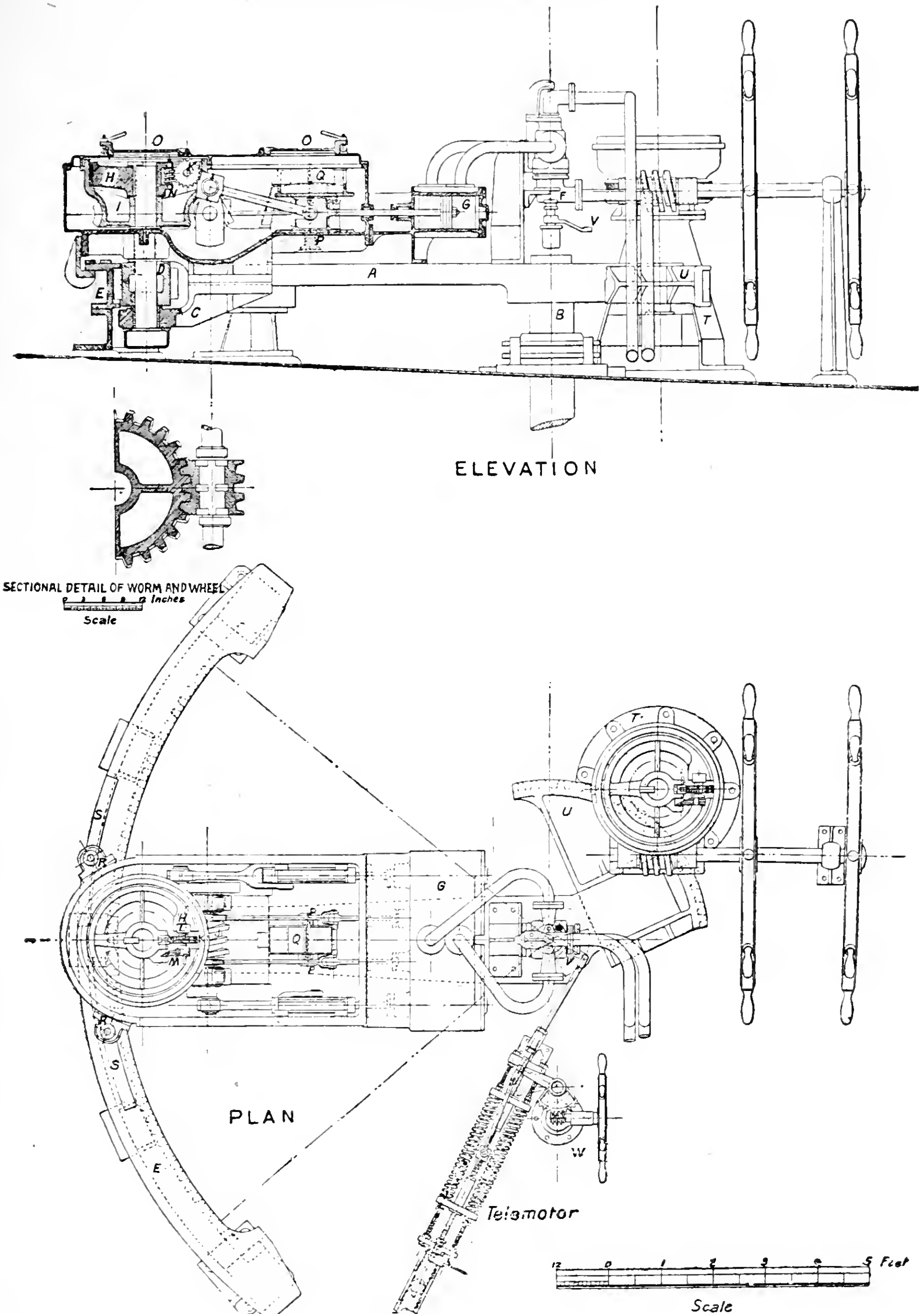


Fig. 195.--Brown's Steam Steering Gear.

largest class of vessels has reached the length of 17 feet. It will be obvious that the strains at the end of such a lever will be reduced to the smallest possible amount, and that the gear necessary to give the requisite power to steering the ship will be of the simplest form.

This tiller is shown in fig. 195, A, keyed to the rudder-head, B, and at the other end a jaw, C, is fitted with gun-metal bearings, into which a driving pinion, D, works, gearing into the toothed segment, E, which is bolted securely to the deck. The steering engines are carried on the tiller and move round with it, receiving and exhausting their steam through a double stuffing-box arrangement, F, which also contains the reversing valve, and is mounted on the axis of the rudder-head. The steam cylinders, G, are of the usual well-known construction, fitted with piston valves. Motion is communicated to the pinion, D, which is of cast steel, through the intervention of an expanding friction clutch, H, which is lined with elm wood, and engages the worm-wheel, I. This wheel, to reduce friction, is carefully machined in the teeth, and made an exact fit to the worm, J, which is of Admiralty gun-metal, and works in the worm-wheel without any back-lash or shake. The worm is also curved to the radius of the wheel so that three teeth are engaged, thus prolonging the life of the gearing by three times that of a straight worm.

Motion is given to this worm by the steam engine as shown. The clutch, H, is expanded by a screw bolt and worm-wheel, K, which turns in and out of the nut, L, at one end, the other abutting against a series of laminated springs, M, so that by turning the worm, N, by a handle (provided for the purpose) to the right or left, the steam gear is engaged or disengaged at any position the rudder may be in, and at the same time it forms an efficient brake to seize hold of the rudder in a sea-way. In practice it is usual to expand this friction brake or clutch sufficiently tight to put the rudder hard over at full-speed trials; but the springs in any case have not sufficient force to hold the connection tight enough to cause fracture of any part of the machinery. In the event of a heavy sea striking the rudder, it immediately slips, allowing the rudder to move out of position; but by that act the steam valve is opened and the engines bring the rudder back to its normal place. As the steam tiller is intended to work (and in most cases has been so fitted) on the open deck, without any house, the whole of the machinery is placed in a water-tight casing, which forms the framework of the steering engines, access to which is got by the doors, O O. The oiling of the various parts is effected automatically by two valveless oil pumps, P P, driven off the valve-rods of the engine. These throw the oil from a well in the bottom of the casing through the hollow piston-rod into the reservoir, Q, and from there a copious supply of oil is supplied to every working part, as well as the piston and valve-rods. In actual practice the oil is renewed once in three months, about 2 gallons being required.

The pinion end of the tiller is carried up by gun-metal slippers and spiral springs under the lugs, R R, which are capable of adjustment. As there is always a tendency of the tiller or quadrant to shake or chatter when there is no strain on, the rudder being fore and aft, the slippers in that position work upon inclined planes, S S, which gives sufficient brake action to prevent any vibration. At the same time, when putting the helm over to such an extent that the pinion bears hard on the rack, the slippers run down on to the flat part of the toothed segment, when the springs slack off, and only carry up the weight of that end of the tiller.

The hand gear consists of a strong standard, T, bolted to the deck, and carrying an exactly similar worm-wheel, and worm with hand-wheels and friction clutch as that described in the steam gear. At the lower end of the shaft there is a similar pinion to D, which engages the toothed segment, U, of cast steel, which is securely bolted to the steam tiller. The operation in changing from hand to steam or steam to hand by means of these clutch brakes can be, and has been performed, without any undue haste, in half a minute. It may here be pointed out that the result of actual experience is that, with this system of hand gear, the friction is one-third of that of the double screw system with nuts and connecting-rods to a cross-head on the rudder-head. Therefore, one man on the worm-wheel gear is as effective as three on the double screws.

The hand wheels, it will be observed, are set to one side of the centre line, which economises space fore and aft, and brings the position of the man steering immediately opposite the compass.

The steering valve is operated by the lever, V, which causes the piston valve to turn on its axis inside the trunnion casing, and as the tiller moves round it carries the valve face with it and so closes the port. The lever, V, is connected to the motor cylinder of the telemotor gear, leading up to the bridge by two pipes $\frac{3}{8}$ inch in diameter. In case of accident to these pipe communications to the bridge, a steering station, W, is shown aft, which can be connected to the steering valve.

It is claimed for this design of steering gear that it has the fewest number of parts possible—namely, one pinion, one worm-wheel and worm—which, it can easily be seen, is due to the fact that the toothed segment represents in a 10-feet tiller a steering wheel 20 feet in diameter, and this rack being shrouded to the points of the teeth and bolted at short intervals to the steel deck, is extremely secure.

From a commercial point of view, there is a distinct saving in the adoption of such a design, as no space is required for a steering engine amidships, and no house is required aft—a few stanchions and rods or netting being placed at the extremity of the gear for the protection of passengers.

Refrigerating Machines, whereby the air in any chamber can be cooled down to any degree of temperature required and kept steadily at that temperature, are now in great demand for the comfort and well-being of passengers on mail steamers, for the safety if not the comfort of those on warships, and as a means of transporting food of all kinds from the ends of the earth to those countries which are unable to produce sufficient for the wants of those dwelling in them. By their means also the luxuries of life are brought cheaply to the doors of even poor people at very moderate cost. The original idea on which such machines were projected was to abstract the heat of air after compression, so that when it was freed again and expanded it robbed its surroundings of the heat necessary for that process. If, therefore, a machine compressed air to, say, 100 lbs. pressure, its latent heat became sensible and great, so that on passing it into a chamber with pipes through which cold water was forced it was cooled down to nearly the temperature of the water; on admitting this air to a store chamber or powder magazine on expansion it would freeze the water particles in its neighbourhood, producing snow, and lowering the temperature of the air in them to that desired. Or if, instead of admitting the free expanding air into the chamber it is used to cool down the water in a system of pipes, like that of a heating apparatus fitted within the chambers, the action will not be so sudden or so spasmodic, and the whole much more under control. To-day other and improved methods are adopted, one of which is that of Messrs. J. & E. Hall, in which carbonic acid gas is used instead of air with advantage, as it can be made and bottled up at comparatively little cost.

Fig. 196 shows the machine made for use on warships in order that the temperature in the magazines containing the modern explosives shall be kept sufficiently low to prevent their decomposition, and the dangers arising from the instability of these compositions when warm. Fig. 197 is an illustration of the larger horizontal type used on merchant ships to chill the air in the cargo holds for the safe conveyance of meat, fruit, vegetable products, etc., which can only be preserved by maintaining them at a low temperature without freezing their fluid portions. This particular machine has an ice-making rating of 30 tons per twenty-four hours.

The machine consists of two complete units, of which the compressors are made from solid blocks of high carbon forged steel, coupled to the tail rods of a compound steam engine, whose crank-shaft is in two parts, with a flywheel mounted on each coupling, and so capable of being disconnected for the independent running of either side when required in an emergency. The CO_2 condenser coils are of solid drawn copper contained in the two halves of the base casting, and are arranged for withdrawal sideways into the space which has to be provided for the gangway. By this arrangement the coils can be exposed for their entire length for cleaning

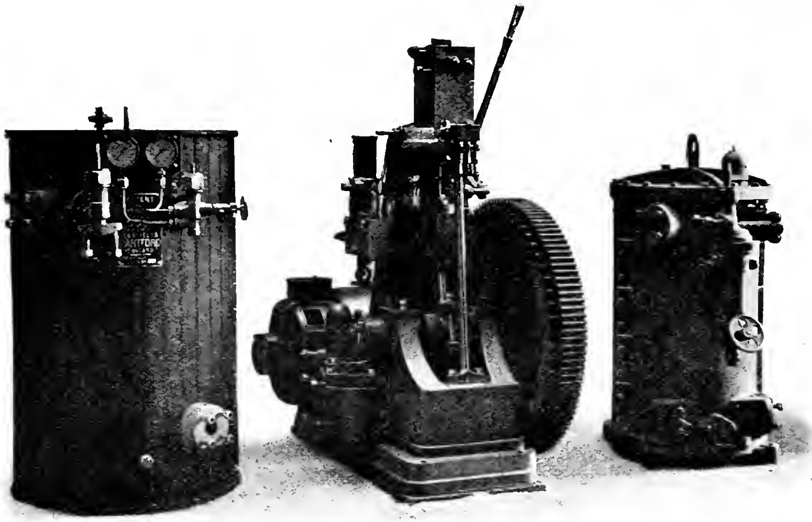


Fig. 196.—Refrigerating Apparatus for Naval Ships.

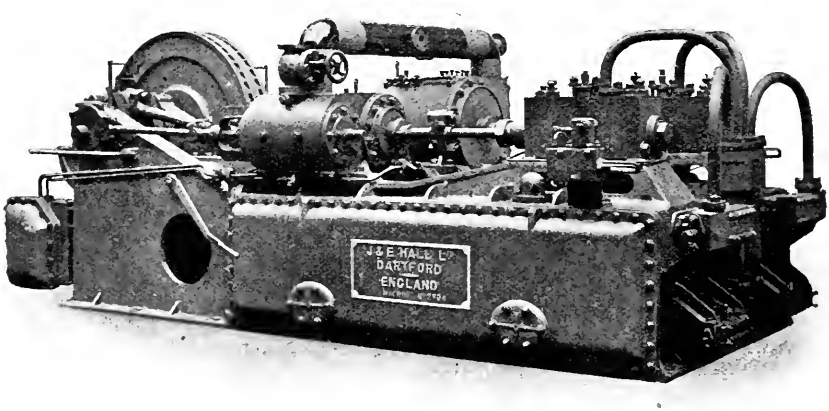


Fig. 197.—Refrigerating Machine for Large Merchant Ships (J. & E. Hall).

when the large door is removed, without disturbance of the gas joints. The evaporator coils, of wrought-iron hydraulic tubing, electrically welded into continuous lengths, are contained in built steel casings constructed to suit the space available in each particular case. The coil boxes, or headers, on the condensers and evaporators are machined from solid mild steel forgings, which also form the stop valve chests, and thereby keep the number of gas joints at a minimum. Each half of the machine is complete with its own pressure lubricator, oil separator, condenser and evaporator gauges, and other accessories.

Fig 198 is a section in diagrammatic form, which shows clearly the working of the Hall system. It will be seen by it, and by reference to fig. 196, that the apparatus supplied to the Admiralty is very compact, and occupies little space, and every part kept small enough to pass through the small opening available in the lower regions of warships. It is worked by an

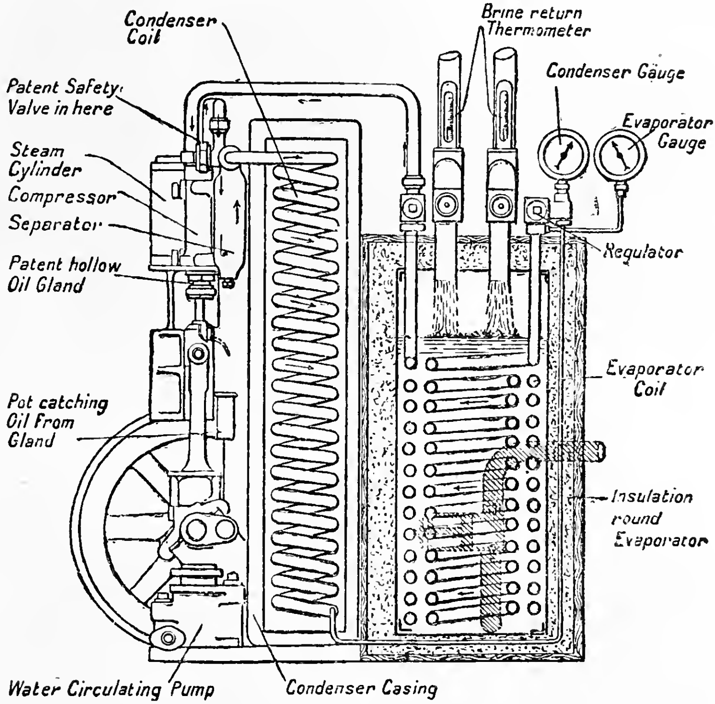


Fig. 198.—Section of Hall's Refrigerating Apparatus.

electric motor through machine-cut wheel gearing, and can be operated easily by the ordinary attendant in the magazine quarters.

Auxiliary Condenser.—In these days of high pressures of steam when fresh water only must be used in steam generators, it is essential to save all the steam possible, and as large quantities are used by the auxiliary machinery in modern steamers, it is worth while having a special condenser for this purpose. In the mercantile marine it is generally called a *winch condenser*, as primarily it was fitted to take the exhaust steam from the winches and return it to the auxiliary or donkey boiler supplying them. These condensers in the Navy are similar in design and construction to the main condenser, but of course much smaller; a combined air and circulating pump is attached and kept working, when in use, at a constant speed; an automatic regulating valve can be fitted in such a way that a the vacuum improves steam is reduced, and when 24 inches of vacuum is obtained, the engine goes "dead" slow.

Fig. 199 gives a simple arrangement for the mercantile marine which

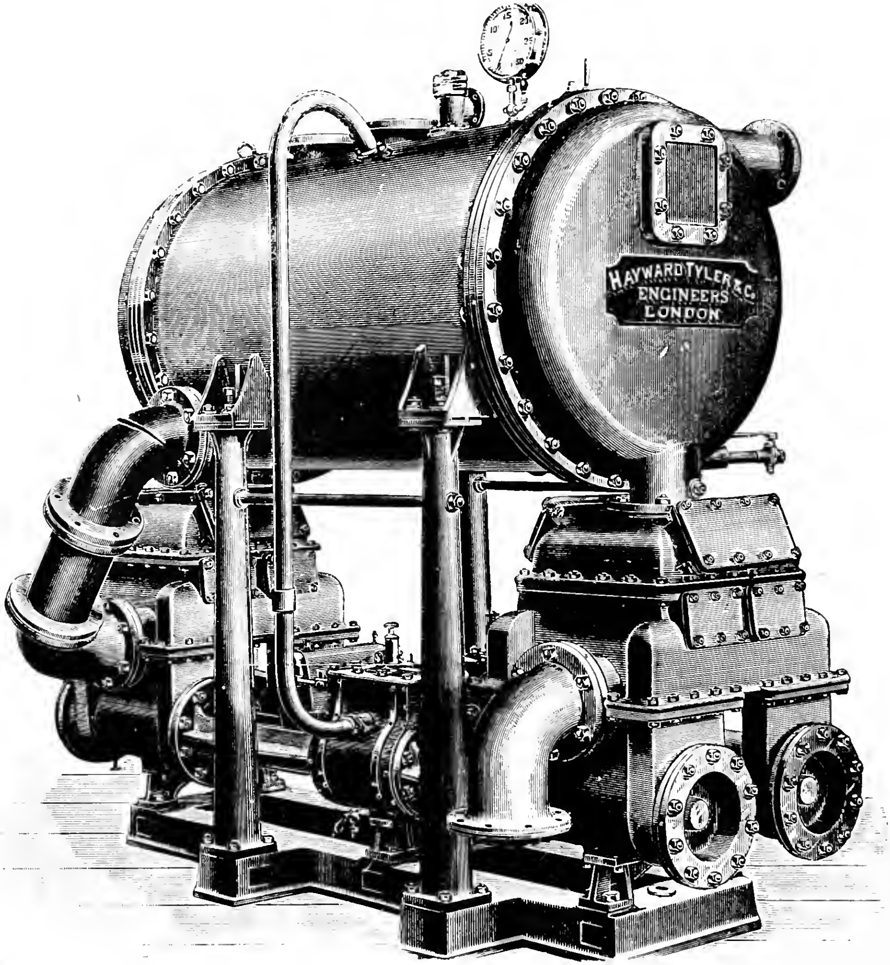


Fig. 199.—Winch Condenser, complete, with Air and Circulating Pumps

may be placed in any convenient position in engine or boiler room, as it is self contained. Sometimes in the mercantile marine a plain cylindrical condenser is fitted so that the ballast pump supplies circulating water, and one of the auxiliary feed-pumps draws away the condensed water; in this case little or no vacuum is maintained, but no water is lost. In every-day work the winch and other auxiliary engine glands and drain cocks leak so badly that only a very poor vacuum is possible with even a good-sized air-pump.

Fig. 200 is a modern high-class surface condensing plant, such as is used on shipboard when a good vacuum is required for working such things as turbine generators, etc. It is also a good representation of a combined set of air-pumps and circulating-pumps, such as may be employed on a

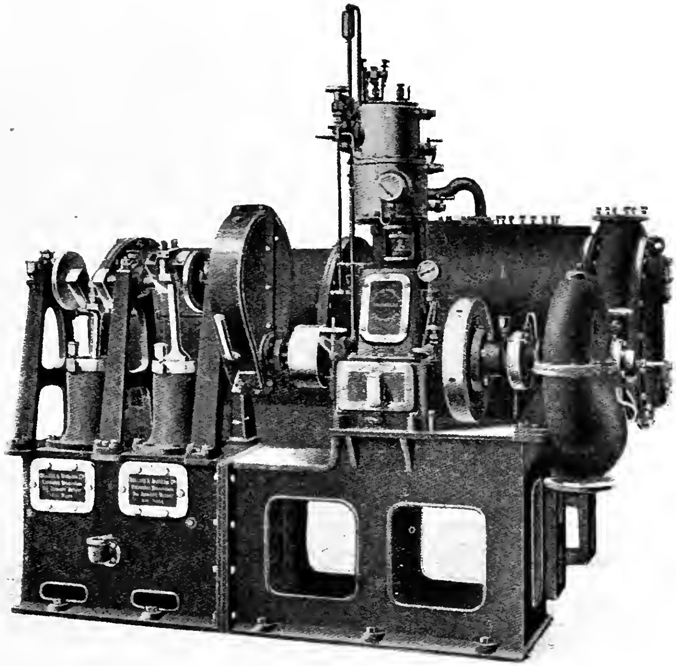


Fig. 200.—Auxiliary Surface Condensing Plant (Belliss & Morcom).

main condenser. It consists of a pair of Edwards' air-pumps worked by cranks driven by a spur-wheel having helical machine-cut teeth, into which there is a pinion on the crank-shaft end of a tandem compound enclosed forced-lubricating engine; to the other end of the crank-shaft the spindle of a centrifugal pump is coupled. In large sizes there are sometimes three pumps driven by cranks at 120° apart, but really the two pumps with cranks opposite one another are sufficient.

CHAPTER XXI.

BOILERS, FUEL, ETC., EVAPORATION.

The boiler proper consists essentially of two parts; the one the fire-place in which the fuel is burnt and heat generated, the other in which the heat is applied to boil the water and convert it into steam.

The part where the fuel is burnt is called the *furnace*, and that on which it is laid is called the *grate*.

The quantity of steam generated will depend, in the first place, on the quality of the fuel, and on the quantity burned; and in the second place on the capacity of the boiler for absorbing and transmitting the heat generated. The quantity of solid fuel consumed depends on the area of grate, and the draught or flow of air into the furnace.

The *efficiency* of the furnace depends on its capability of burning with as little waste as possible the whole of the fuel in it, and that without superfluity of air; also, to produce perfect combustion there must be as little waste of heat as possible in obtaining the necessary draught. The portion of heat generated which is applied to the production of steam is called the *available heat*.

Experience has proved that a grate may consume a large quantity of fuel without thoroughly burning it, and that even when the fuel is thoroughly burnt only a comparatively small portion of its heat may be usefully employed.

The chief loss is at the chimney, which is a rough and ready way of inducing the air to flow into the furnaces with sufficient velocity to cause the fuel to burn; but it is an exceedingly wasteful one, and will, some day, undoubtedly be superseded by a more scientific and economical apparatus. One pound of fairly good coal *can be made* to evaporate 14 to 15 lbs. of water; but in the best of marine boilers only 10 to 11 lbs. of water per pound of such coal are evaporated in practice with every care taken, showing that the efficiency of the boiler is less than 0·75 from this point of view. With picked Welsh coal and most careful stoking 12 lbs. can be evaporated if the rate of combustion and evaporation is not high.

Efficiency of the Furnace.—Loss may take place at this part of the boiler from the following causes:—

(1) *Bad stoking*, whereby fuel is lost by falling through the bars only partly consumed, and is thrown away with the ashes. This generally takes place with good fuel from its friability; but may be due in some cases to carelessness in laying the fire-bars. Unevenly disposing of fresh fuel and allowing clinkers to form are also causes of inefficiency. Too much cannot be said of bad stoking, as from this cause alone the best-designed and made boiler may prove most inefficient.

(2) *Want of air*, whereby the fuel is not wholly consumed, but part of it

passes off through the funnel as carbon monoxide, part in the form of smoke, part is deposited as soot in the tubes, and part is burnt in the smoke-box. This is sometimes due to bad design and want of proper means of regulating the supply of air; but it is more frequently due to bad stoking and carelessness on the part of the fireman in using the means provided.

(3) *Excess of air*, whereby some of the heat is employed in raising the temperature of the superfluous air, and thereby doing no good. This is generally caused by bad stoking, the bars being uncovered in parts of the grate, and so admitting a too large inflow of air.

(4) *Radiation* through the mouth when the doors are open for firing, and any radiation through openings for other purposes. This cannot be avoided, but should be comparatively small.

The first three of these sources of loss cannot be said to be unavoidable, as with ordinary care the loss from them should be very small compared with the chief loss.

Chimney Draught.—To obtain an efficient draught in a chimney or funnel, the temperature at its base should be about that of melting lead, or nearly 600°. Professor Rankine said that "the best chimney draught takes place when the *absolute* temperature of the gas in the chimney is to that of the external air as 25 to 12."

That is, if T be the temperature of the air, then

$$\text{Absolute temperature at the base of funnel} = \tfrac{25}{12} \times (461^\circ + \text{T}),$$

or temperature according to thermometer (Fahrenheit)

$$= \tfrac{25}{12}\,(461^\circ + \text{T}) - 461^\circ = 2{\cdot}08\,\text{T} + 500^\circ.$$

Taking the air temperature at 60°, the proper heat at the base of the funnel is $2{\cdot}08 \times 60 + 500$, or 625°. Now, the heat in a furnace having a natural draught is usually about 2,400°, and if the heat at the exit from the boiler to obtain the necessary draught is 600°, it follows that 20 per cent. of the total heat of combustion is wasted owing to this. If, instead of a chimney, a draught were produced artificially, say by means of an economic engine driving a fan, a very considerable portion would be saved.

Fuel.—The solid fuels used on board ship are *coal* of various descriptions and qualities, *wood* in those parts of the world where it is plentiful and coal scarcer, and *patent fuels* or combinations of very small coal with other substances. *Coke* is seldom used, but *oil*, comparatively cheap, and in some instances most convenient, has now come into general use, especially in warships. Mineral oil is capable of producing a large quantity of heat, is clean and convenient for stowing, and therefore suitable for warships, yachts, and passenger steamers; arrangements are now made for the regular supply of a high flash point cheap oil at foreign and home ports, so that ships can depend on getting their tanks filled as surely as they can their bunkers. Creosote waste, as well as petroleum refuse, can be burnt quite safely and readily in the furnaces of a boiler by means of special apparatus; the one invented by Mr. James Holden, and used by him so successfully for many years in the locomotives of the G.E.R. Co., London, is shown in detail in fig. 201, and fig. 202 gives a very simple form used on the Japanese ships. Residue, after the light illuminating oils have been abstracted from

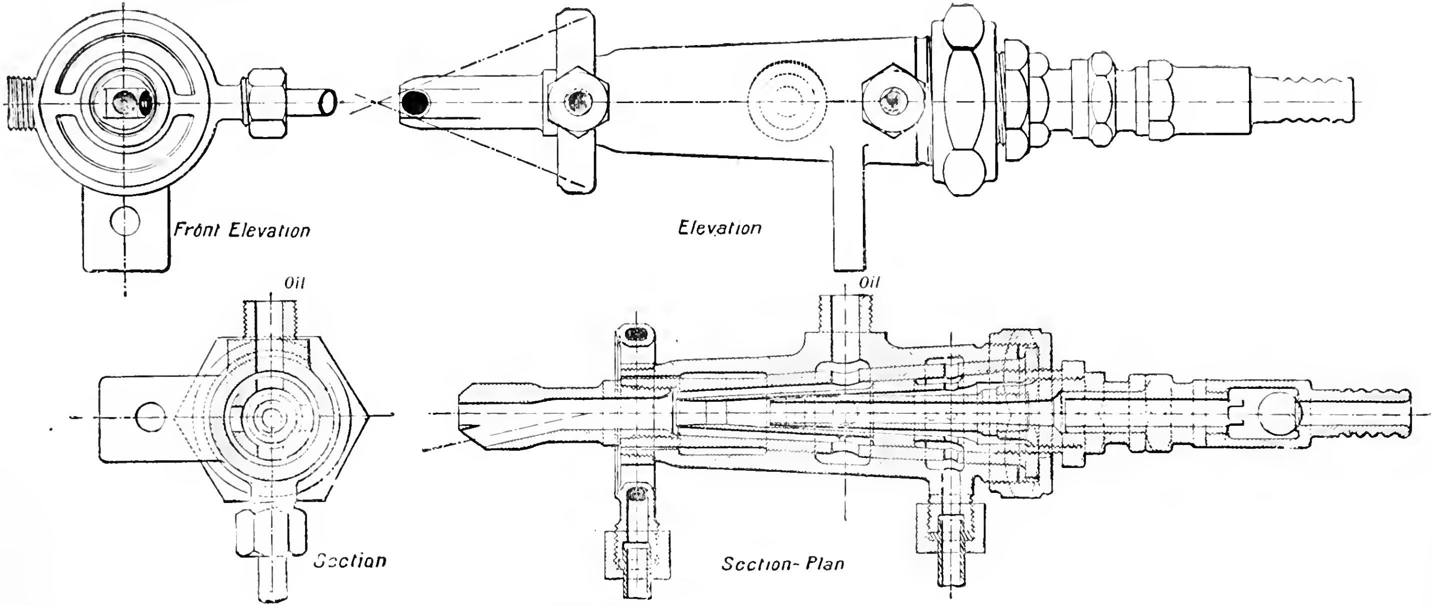


Fig. 201.—Holden's Apparatus for using Oil as Fuel in Locomotives.

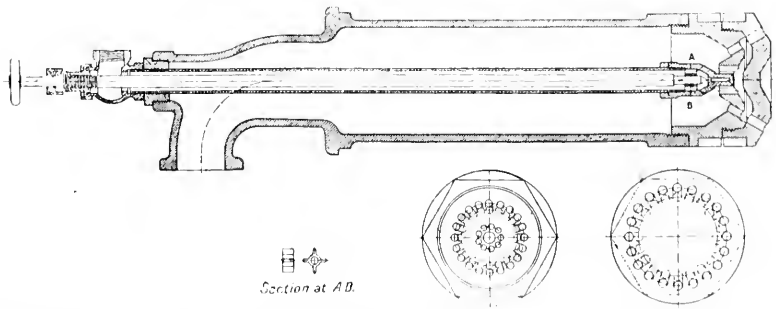


Fig. 201*a*.—Lassoe-Lovekin Burner for Atomising Heavy Oil with Air.

the petroleum, in South Russia called *astatki*, has for very many years been used on the steamers of the Caspian and Volga for fuel.

Of coals there are several distinct kinds, and many more qualities. There are five distinct varieties, known as—

(1) *Anthracite*, consisting almost entirely of free carbon, generally jet black in appearance, but sometimes greyish like black lead, has a specific gravity generally of about 1·5, but sometimes as high as 1·9; it burns without emitting flame or smoke, but requires a strong draught to burn at all. It is capable of evaporating (theoretically) nearly 16 times its weight of water, but to obtain good results from it careful stoking is necessary, as when suddenly exposed to heat it is very friable, breaks up into small pieces, and falls through the bar-spaces if disturbed much, as it does not cake. The fires should be worked light when using it, and the coal carefully spread. The heat is very intense and local, so that furnaces intended to burn it should be high in the crowns and protected at the sides by bricks, fireclay, etc., or else have no air-spaces down the sides.

(2) *Dry bituminous coal* contains from 70 to 80 per cent. of carbon, and about 15 per cent. of volatilisable matter; its specific gravity is from 1·3 to 1·45. It burns easily and swells considerably while being converted into coke. The harder kinds do not burn so readily, nor do the pieces stick together so easily when burning, and are generally better adapted for marine boilers.

(3) *Bituminous caking coal*, containing from 50 to 60 per cent. of carbon, is generally of about the same specific gravity as the dry bituminous; it contains, however, as much as 30 per cent. of volatilisable matter, and consequently develops hydrocarbon gases; it burns with a long flame, and sticks together in caking, so as to lose all trace of the original forms of the pieces. It requires special means to prevent smoke.

(4) *Cannel coal*, or *long flaming coal*.—This is seldom used for steam purposes, as it gives off large quantities of smoke, and is very scarce. It is the best coal for the manufacture of gas.

(5) *Lignite*, or *brown coal*, is of later formation than the other coals, and in some instances approaches to a peaty nature. It contains, however, when good, from 56 to 76 per cent. of carbon, and has a specific gravity from 1·20 to 1·35. It also contains large quantities of oxygen, and a small quantity of hydrogen. The commoner kinds of lignite are poor, and contain as little as 27 per cent. of carbon, and, therefore, are not suitable for steaming purposes.

In some inland parts of the world, where coal cannot be easily obtained, wood is used for fuel, and the furnace specially constructed to burn it; it contains, on the average, when dry, about 50 per cent. of carbon, 41 of oxygen, and 6 of hydrogen.

Patent Fuels consist principally of coal dust, and their value depends, therefore, on the quality of the coal from which they are made. To utilise the small coal from the mines and yards was a somewhat difficult problem, as it could not be conveniently transported, and is difficult to burn in an ordinary furnace. Friable coal is also equally difficult to deal with. By mixing a small quantity of pitch or tar with it, and baking the mixture in moulds, a hard brick is produced, which is easily handled and burns well.

There are many other ways of manufacturing patent fuel from small coal, but the result is the same—viz , a hard compact brick.

Oil Fuel is now largely used in warships and in many of the mercantile ships in various parts of the world, where it can be obtained in quantity

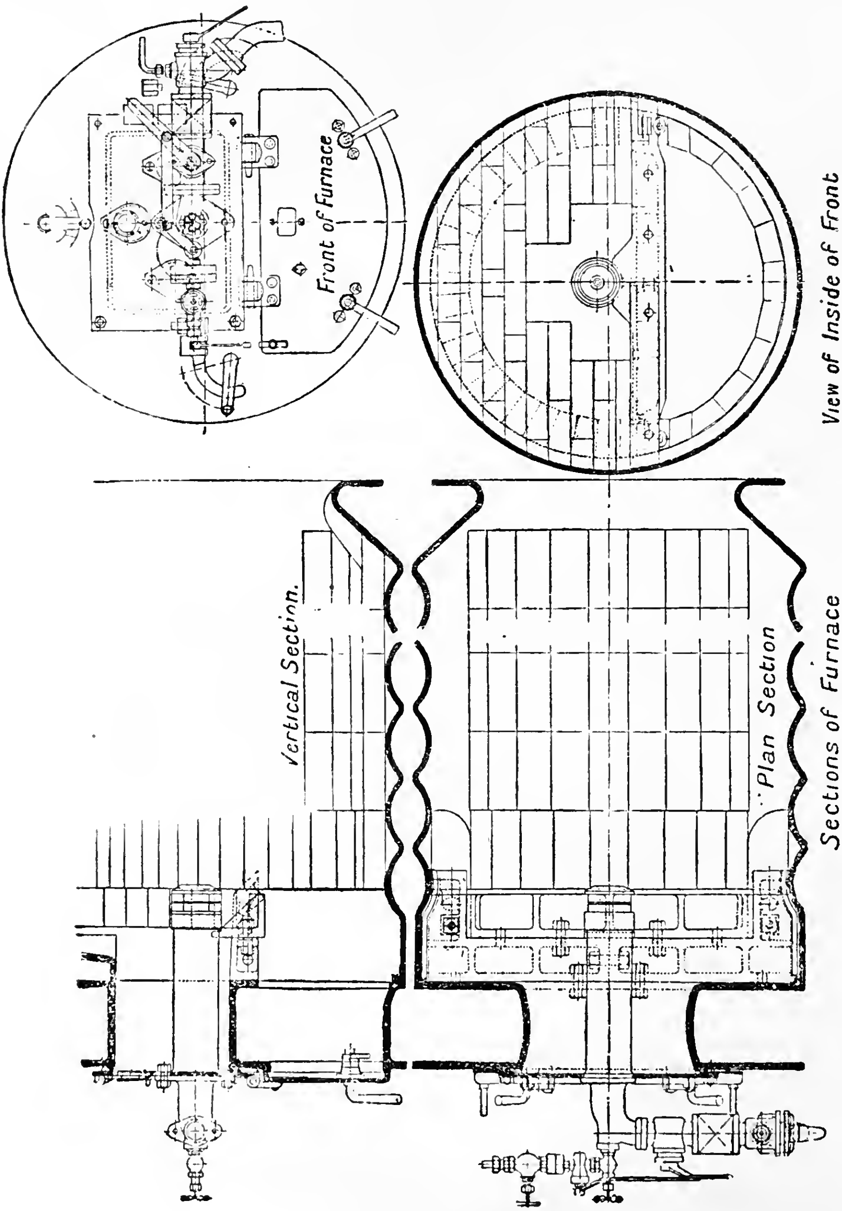


Fig. 202.—Burner Gear for using Heavy Crude Oil as Fuel.

cheaply. It is used chiefly at present for raising steam, and, therefore, its composition is of small moment compared with its calorific value, but in the near future the demand will be considerable for use in the oil internal-combustion engine, when the composition will be a matter of supreme importance, inasmuch as those oils containing asphaltum matter in appreciable quantity cannot be used in spray form in an engine cylinder, owing to the deposition of coke sand of a hard refractory nature, which will not consume nor freely depart from within, consequently the pistons, cylinder walls, rods, etc., get badly damaged. To use such oils in an internal-combustion engine successfully the pitchy matter must be deposited outside, and only the volatile constituents permitted to enter the cylinders.

Crude Oils, pumped from the wells, after exposure, can be used in boilers for raising steam, and, being cheap and easily handled on shipboard, have great advantages over coal, especially where coal is imported and comparatively dear. Such oil, however, contains much volatile matter, and is, therefore, not so safe as coal, nor so safe as the residue from it after refining, which is quite as good for burning, and has a high flash point.

Table li*a*. gives the composition and heat value of all the principal oils now used, as well as that of the residues and of coal tar.

Admiralty Conditions for Oil Fuel.—The *flash point* is not to be below 175° F. *The sulphur contents* are not to exceed 3 per cent. Practically *free from acid*, which must not exceed 0·05 per cent. *The water* with it must not exceed 0·5 per cent. *The viscosity* such that the flow at a temperature of 32° F. by the Redwood viscometer does not exceed 2,000 seconds for 50 cubic centimetres. There must be *no earthy or fibrous matter* with it.

The other Heavy Oils used for the Diesel and other engines are the residues after the light volatile and inflammable constituents, such as petrol, benzine, paraffin, etc., have been extracted, and the flash point is then over 200° F., and often as high as 250°. The Caspian astatki has a higher flash point, being generally as high as 390° F. American residues are those remaining after lubricating as well as lighting oils have been distilled.

Oil Burners for Marine Boilers are designed to pulverise the oil as it flows through their nozzles, and to direct the spray and the consequent flame to that part of the furnace desired. There are three ways of effecting this—(1) By means of a steam jet, (2) by an air stream, and (3) by purely mechanical means. The steam injector is the simplest, and has been used with success by Mr. Holden and others on locomotives, but on shipboard the loss of water is too serious a matter, and so some marine engineers prefer the air blast, which at comparatively low pressures is much more economical than the steam jet; at high pressures, however, the air system is very little, if at all, superior to the mechanical methods. The mechanical methods do not involve so great a steam consumption nor such expensive mechanism as air compressors, nor do they require quite so much space. The oil is by them forced into the burners at pressures varying from 50 to 250 lbs. per square inch, and there heated by steam to temperatures varying from 150° to 300° F. The fires thus formed may be under natural or forced draught, even to such a forcing as measured by 5 inches of water, but the best results are obtained at quite low rates of combustion, when with decent oil 14 to 16 lbs. of water (from and at 212° F.) can be evaporated per pound of oil, as against 8 to 11 lbs. when using best Welsh steam coal.

TABLE LI.—FUELS.

Description.	Carbon.	Hydrogen.	Oxygen.	Sulphur.	Ash, &c., including Nitrogen.	B.T.U. per Lb. Fuel.	Water Evaporated per Lb. Fuel from 212°. B.T.U. ÷ 966.	One Ton occupies in Cubic Feet.
							Pounds.	
Welsh—Ebbw Vale,	87·78	5·15	0·39	1·02	3·66	16,221	16·79	..
,, Powell's Duffryn,	88·26	4·66	0·60	1·77	4·71	15,788	16·34	..
,, Llangennech,	84·97	4·26	3·50	0·42	6·85	14,682	15·20	..
,, Græigola,	84·87	3·84	7·19	0·45	1·91	14,130	14·63	..
,, Average (38 samples),	83·87	4·79	4·15	1·43	5·89	14,858	15·52	42·7
Newcastle Average (18 samples),	82·12	5·31	5·69	1·24	5·12	14,820	15·32	45·3
Derbyshire Average (7 samples),	79·68	4·94	10·28	1·01	4·06	13,860	14·34	47·4
South Yorkshire,	81·88	4·83	7·47	0·54	2·95	14,296	14·71	46·0
Lancashire Average (28 samples),	77·90	5·32	9·53	1·44	6·18	13,918	14·56	45·2
Scotch Average (8 samples),	78·53	5·61	9·69	1·11	5·03	14,164	14·65	42·0
Irish Anthracite,	80·03	2·30	..	6·76	11·03	13,302	14·5	35·7
American Anthracite, Best,	88·54	2·43	..	0·04	8·60	15,163	15·67	42·35
,, Pocahantos,	85·94	4·45	4·50	0·82	4·29	14,990	15·50	42·44
French Anthracite,	86·17	2·67	2·85	..	8·56	14,038	14·53	40·00
,, Hard Bituminous,	88·56	4·88	4·38	..	2·19	15,525	16·10	42·75
,, Caking ,,	87·73	5·08	5·65	..	1·54	15,422	16·00	42·75
Chilian Coal,	63·56	5·43	14·84	2·50	11·13	11,030	11·68	..
Indian Coal, Average,	70·20	..	..	..	22·9	..	..	..
Patent Fuel—Warlichs,	90·02	5·56	..	1·62	..	16,495	17·07	..
,, Average,	83·40	4·97	2·79	1·26	6·01	15,000	15·66	34·4
Lignite—Russian,	73·72	6·09	20·19	ash neglected		14,263	14·7	..
Kentish Coal,	79·78	..	16·73	0·8	2·08	13,900	14·4	..
Coke—Best Durham,	85 to 92	..	..	0·25 to 2·0	4 to 12	12,832	13·30	..
Woods—Beech,	49·36	6·01	42·69	..	1·91			94·4
Oak,	49·64	5·92	41·16	..	3·26	average		94·4
Birch,	50·20	6·20	41·62	..	1·96	when dry		106·1
Fir,	51·79	6·28	41·93	..	..	7,800	8·1	..
Willow,	49·96	5·96	39·56	..	4·33	..	..	124·7
Peat—Fairly dry,	59·6	5·80	29·6	0·3	4·7	9,951	10·30	..
Petroleum—American crude,	84·7	13·1	2·2	1·63 to 2·43	..	20,240	20·95	39 to 46

N.B.—The available production in practice is about 77 per cent. of the above with good natural draught, and 70 per cent when rate of evaporation is high per square foot T.H.S.

TABLE LIa.—Liquid Fuels, Composition and Calorific Values of.

N.B.—Actual Evaporation in Practice is about 70 per cent. when highly forced, and 77 per cent. when lightly forced.

Description and Origin.		Flash Point.	Specific Gravity.	Chemical Components.				B.T.U. per Lb. of Fuel.	Evaporation per Lb. from and at 212°. B.T.U. ÷ 966.
				Carbon.	Hydrogen.	Oxygen.	Sulphur.		
		F.°							
Heavy Petroleum,	W. Virginia,	..	0·873	83·50	13·30	3·20	..	18,324	18·97
,, ,,	Pennsylvania,	..	0·886	84·90	13·70	1·00	..	19,210	19·88
Crude ,,	Mexico,	..	0·937	83·21	11·32	2·90	2·43	17,460	18·07
,, ,,	Texas,	180°	0·924	84·60	10·90	2·87	1·63	19,060	19·73
Treated ,,	,,	216°	0·926	83·26	12·41	3·83	0·50	19,480	20·16
Residue ,,	California,	278°	0·864	84·00	11·07	3·93	0·87	17,400	18·01
,, ,,	,,	185°	0·935	84·78	11·85	2·45	0·84	17,550	18·17
Treated ,,	,,	311°	0·966	81·52	11·01	NO 6·9	0·55	18,667	19·32
Petroleum from	Parma,	..	0·786	84·00	13·40	1·80	..	18,218	18·96
,,	Pechelbronn,	..	0·892	85·70	12·00	2·30	..	18,036	18·67
,,	E. Galicia,	..	0·870	82·20	12·10	5·70	..	18,153	18·79
,,	W. Galicia,	..	0·885	85·30	12·60	2·10	..	18,416	19·06
,, (Residue)	Roumania,	284°	0·930	..	..	0·43	0·20	18,900	19·56
,, ,,	Balakhany,	..	0·882	87·40	12·50	0·10	..	21,060	21·80
,, (Light)	Baku,	..	0·884	86·30	13·60	0·10	..	20,628	21·35
,, (Heavy)	,,	..	0·938	86·60	12·30	1·10	..	19,440	20·12
,, (Residue)	,,	..	0·928	87·10	11·70	1·20	..	19,260	19·93
,, ,,	Java,	..	0·923	87·10	12·00	0·90	..	19,496	20·18
Shale Oil,	France,	..	0·911	80·30	11·50	NO 8·7	..	16,283	17·41
,,	Scotland,	..	0·862	85·35	12·44	..	0·29	18,317	18·96
Coal-tar oil (light),	British,	120°	0·958	86·16	9·05	..	0·30	18,062	18·70
,, (heavy),	British,	..	1·038	84·50	7·62	6·28	0·19	16,325	16·90
,, ,,	France,	..	1·044	82·00	7·60	10·40	..	16,050	16·61
,, ,,	Germany,	208°	1·065	90·12	7·09	1·66	0·63	15,950	16·51
Gas oil (light),	British,	176°	0·850	86·98	12·15	0·16	0·71	18,000	18·63
,, (heavy),	,,	..	1·067	87·62	5·95	..	0·67	16,153	16·72

N.B.—For evaporation per gallon multiply by (10 × specific gravity).

The Value of a Fuel is determined by its chemical composition. All fuels contain more or less carbon, and must have hydrogen and oxygen in various proportions, also some small quantities of nitrogen, sulphur, etc.

These substances are usually designated by their chemical symbols, that is by the initial letter of the name. They combine in certain fixed quantities, called their *chemical equivalents*. Thus—

Carbon,	symbol C,	chemical equivalent	12.
Hydrogen,	,, H,	,, ,,	1.
Oxygen,	,, O,	,, ,,	16.
Nitrogen,	,, N,	,, ,,	14.
Sulphur,	,, S,	,, ,,	32.

A pound of carbon is capable of developing, during combustion, a certain quantity of heat, called the *total heat of combustion*, and is measured by *units of heat*.

The British standard * unit of heat is defined as *that quantity of heat which will raise one pound of pure water one degree Fahrenheit in temperature.*

* Joules found the thermal unit to be equal to 772 foot-pounds; recently, however, with better appliances and possibly greater care in manipulation, Profs. Rowald, Griffiths, and Schuster find it to be 778.

The total heat of combustion of one pound of hydrogen is measured in the same way, and it is also found that the *total heat of combustion of any compound of carbon and hydrogen, is the sum of the quantities of heat which the hydrogen and carbon contained in it would produce if burnt separately.*

If a fuel contains oxygen as well as hydrogen, it is known that eight parts by weight of the former unite with one of the latter to form water, which exists as such in the fuel, and *does not add to the total heat of combustion.* If there is, however, an excess of hydrogen beyond what is required to form with the oxygen the water, the *remaining hydrogen does add to the total heat of combustion,* and may be reckoned in estimating its value.

Hydrogen gas requires 8 pounds of oxygen, and consequently 36 pounds of air to consume it; its total heat of combustion is 62,032 units of heat, and, therefore, it can evaporate 64 pounds of water from and at 212°.

Carbon, when fully burnt, requires only 2·7 pounds of oxygen, or 12 pounds of air to consume it—that is, to convert it into carbonic acid: its total heat of combustion is 14,500 units of heat, and can, therefore, evaporate 15 pounds of water from and at 212°. If, however, it is only partially burnt or turned merely into carbonic oxide, half the quantity of air is consumed, and the total heat of combustion is only 4,400 units of heat.

Sulphur exists only in small quantities in good coal, and the total heat of combustion is only about 4,000 units.

From this information, the following rule is deduced for the total heat of combustion for substances containing carbon, hydrogen, and oxygen.

Total heat of combustion of 1 pound of fuel

$$= 14{,}500\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}, \quad . \quad . \quad . \quad (1$$

and

Theoretical evaporative power of 1 pound of fuel

$$= 15\left\{C + 4{\cdot}28\left(H - \frac{O}{8}\right)\right\}.$$

C being the weight of carbon, H that of hydrogen, and O that of oxygen, all expressed in fractions of a pound.

Example.—To find the evaporative power of a pound of Welsh coal, whose constituents are, carbon 85, hydrogen 4, oxygen 2·5, others 8·5.

$$E = 15\left\{0{\cdot}85 + 4{\cdot}28\left(0{\cdot}04 - \frac{0{\cdot}025}{8}\right)\right\}$$
$$= 15{\cdot}125 \text{ pounds.}$$

The quantity of air required to burn a pound of fuel may also be estimated in a somewhat similar way.

Weight of air required to *burn* 1 pound of fuel

$$= 12\,C + 36\left(H - \frac{O}{8}\right).$$

To *burn* ordinary coal or coke, 12 pounds of air are required on the average.

To provide for the *dilution* of the gaseous products so that free access is given to the air to reach the fuel, much more is required; so that *to consume one pound of fuel in practice, 24 pounds of air are often used.*

At the temperature of 62°, the volume of 1 pound of air is 13·14 cubic feet; therefore, to consume 1 pound of coal or coke 315 cubic feet of air are necessary. If, however, the draught is very good, such as found with artificial means, 250 feet are sufficient.*

Sir Alex. Kennedy found that the best results were obtained when 18·14 pounds of air were used, and noted the following:—

Air per pound of coal, . . . lbs.,	14·5	17·4	17·8	18·14
Loss per cent., ...	9·2	5·0	3·6	0·5

Tables li. and li*a*. give the composition, total heat of combustion, and evaporative power of the various fuels.

Rate of Combustion.—The quantity of coal burnt on a square foot of grate depends partly on its nature, but mostly on the draught. Some hard coals, like anthracite, require a very strong draught to burn at all, and some qualities of anthracite burn slowly even then. Bituminous coal burns much more freely than anthracite, and some of the softer kinds consume very rapidly. All coal burns very much more rapidly with a strong draught, as might be supposed, and for that reason, when only a comparatively small boiler can be fitted to supply steam for the small grate possible with it, artificial draught is a necessity. It is also claimed that when the draught is very strong, *a smaller weight of air suffices* to complete combustion.

Artificial Draught.—There are four general methods of causing an artificial draught: (1) The steam blast, by which the products of combustion are ejected from the funnel in the same way that the exhaust steam from a locomotive produces a draught; (2) by an air-blast delivered under the fires with a closed ashpit; (3) by making the boiler room air-tight, and forcing air into it by a fan, until the pressure is above that of the atmosphere, the only vent being through the furnaces to the funnel; and, (4), by a fan placed near the base of the funnel which draws from the smoke-box and delivers into the funnel; this latter is known as the *induced draught system.*

The first of these methods is the older and costlier form; it is not very effective, and by no means economical, as there is loss of both coal and fresh water; it also quickly wears out the funnel; but it has the merit of being cheap in first cost, and does well enough if it is only required to be used occasionally to quicken the fires in getting up steam, or during a hot day when steaming with a fair wind, or no wind at all. The second plan has the merit of simplicity, and made by the late Mr. James Howden and others very effective by their special systems, it is now very generally used with most satisfactory results in the mercantile marine. There is no danger with it, and it may be applied to any kind of ship. The third plan is one almost universally adopted in warships, and also in many large and small express merchant ships. It is a costly plan, and necessitates a complete change in the ship arrangements in the boiler compartments. It is undoubtedly a most efficient plan of artificial draught, but the cost must enter into

* *The temperature of ignition* of Scotch coal is 760° F., of Tyneside 770°, Durham 790°, Yorkshire 810°, Welsh 875°, and anthracite 925°.

the calculation of *practical* efficiency, as must also the risk run. The firemen are practically imprisoned in the stokehole, as access to, and egress from it are through an air-tight lock, and when a panic arises the result under such

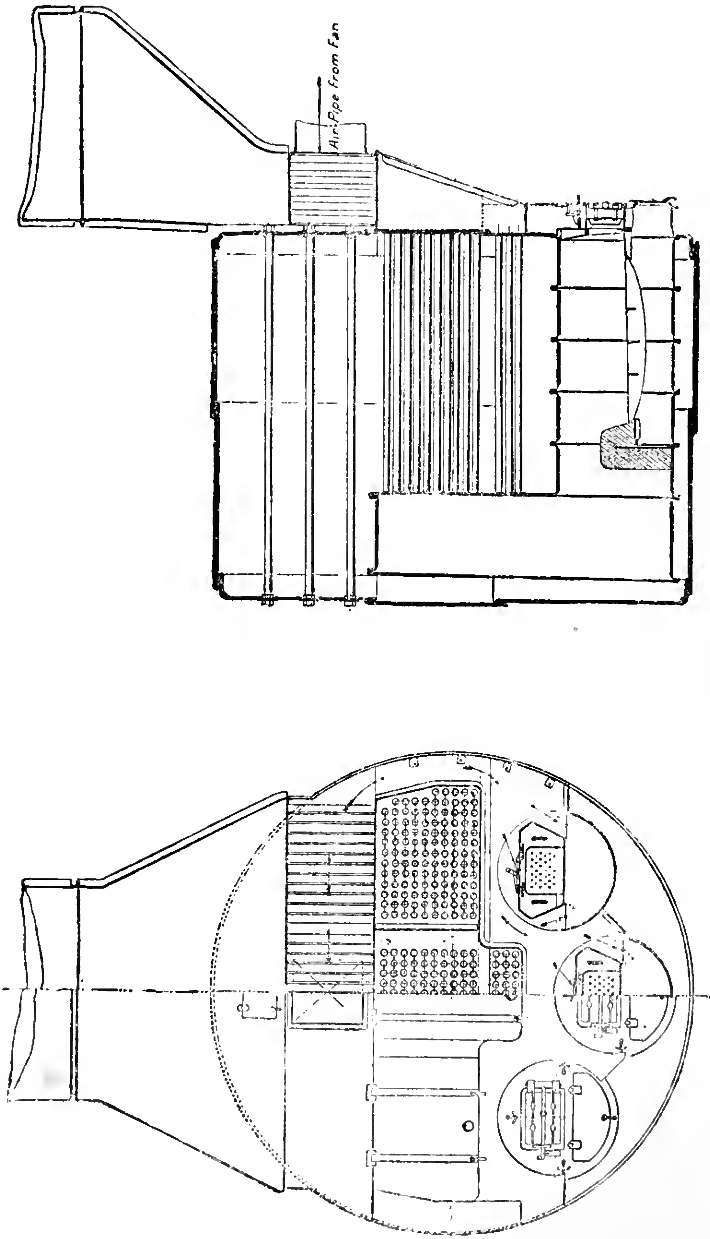


Fig. 203.—Howden's System of Forced Draught.

circumstances might be serious; on the other hand, the fireman is always working in fresh, cool air, and, therefore, more comfortable and efficient than in the ordinary stokehole. The fourth plan is simple and less costly, and was claimed to be less trying to the tubes, etc., but why is not obvious. The Ellis-Eaves patent system, which includes such an eduction fan, has been tried with success; but the fan must be most carefully designed and fitted to enable it to deal with gases at 600° F. successfully.

Howden's System of Forced Draught.—The second method of artificial draught above mentioned was elaborated and perfected by the late Mr. James Howden, who was one of the first to take up the question of forced draught, and to persevere both in experimenting and in educating the public mind, so that to-day his system is very extensively adopted in the Mercantile Marine of all nations with more or less beneficial results. Fig. 203 shows his arrangement whereby the air is supplied to the furnaces by conduits at the sides of the smoke-boxes after having passed through a chamber containing a system of vertical tubes through which the hot air and gases from the smoke-box flow to the funnel. The supply of air is kept up and regulated by means of a fan, and is usually delivered at a pressure equal to half an inch to one inch of water. This system is, therefore, on the regenerative principle, whereby the waste heat in the chimney is utilised to heat the in-coming air to the furnaces, and although the direct saving by using this waste heat is not very large, owing chiefly to the low specific heat of air, the indirect one is considerable from the fact that the combustion of fuel is much more complete and efficient the nearer the air supply approaches the temperature of combustion. Mr. Howden claimed a saving of coal by this system amounting to a considerable percentage, and no doubt this was so when applied to a boiler whose design and proportions were not of the best, but in every case there must be a material gain by supplying the air raised by waste heat at a high temperature * at a rate to suit the conditions prevailing at the time.

Ellis & Eaves' System differs from the Howden system chiefly from the fact that the products of combustion are drawn at the funnel base and delivered up to the funnel by a fan after passing through the tubes of a heating chamber similar to Howden's. In this case the inflow of air is through the heating chamber and downwards to the furnaces, induced by the suction of the fan, and hence called the "induced draught" system. This system also differs from Howden's in some details. Among others, it was usually fitted with boilers having Serve tubes of considerable diameter, as against the tubes of small diameter fitted with retarders in Howden's case. This system was fitted by Messrs. John Brown & Co., of Sheffield, to several ships, as well as to boiler installations on shore, with beneficial results in all cases. It need hardly be said, however, that the suction fan had to be very carefully designed and fitted to avoid damage from the heat of the gases that pass through it.

Quantity of Fuel Burnt on the Grate.—With good stoking and the ordinary funnel draught, as much as 20 pounds of coal can be burnt per square foot of grate per hour, and under most favourable circumstances with short grates 22 to 25 pounds. In the mercantile marine, 15 pounds is the average amount burnt when working economically, and all calculations for a merchant ship should be based on this; for although 20 pounds

* 300° F is often attained when working at full power with a pressure in the ashpits from ½ to 1½ inches of water.

can be *consumed* when the fires are forced, or on a very windy day when the draught is good, no grate of ordinary length should be supplied with more than 15 pounds to obtain complete combustion, and this is all that will be *consumed* on a sultry day. On a short grate, say 1·3 to 1·5 the diameter of the furnace in length, 20 pounds of coal may, however, be burnt economically. With moderate forced draught, 20 to 30 pounds can be readily burnt on a square foot of grate, and 30 to 50 pounds with good air pressure either *forced* or *induced*, and with short grates and good stoking even as much as 60 pounds. A locomotive with a draught produced by the exhaust steam can burn, as a rule, 65 pounds of coal per square foot of grate; and the modern express engine burns as much as 80 pounds.

The coal consumed in the old torpedo boats with the closed stokehole, and a pressure of air *in the stokehole* above that of the atmosphere equal to 6 inches of water, was as much as 96 pounds per square foot of grate, and was even 62 pounds at a pressure of 3 inches only. Mr. Thornycroft eclipsed even this in the yacht "Gitana," where 138 pounds of coal were burnt per square foot of grate; this is exceedingly high, and is only such as can be obtained under the most favourable conditions with every care taken in the ventilating arrangements; the evaporative power under such conditions is, of course, by no means high, although, perhaps, higher than could have been anticipated. In the Navy an air pressure of ½ inch is allowed to count as natural draught, and 2 inches is the limit allowed for forced draught in large ships with tank boilers. With those pressures the combustion is very good and evaporative power very fair.

Size of Funnel.—The natural draught, or that obtained by means of a funnel, is very much influenced by the area of its transverse section, and by the height or distance of its top from the level of the fire-bars. The draught in a funnel is due to the difference in density of the column of gas in it from that of the surrounding atmosphere; the contents of the funnel rise upwards, on the same principle that a bubble of air rises through water. The density depends on the temperature of the gases at the funnel base, and for this reason a good draught cannot be obtained without a comparatively high temperature. The good draught observable on a windy day is due to the tendency to form a vacuum at the mouth of the funnel, by the wind blowing across it. Professor Rankine has given the following formulæ for chimney draught:—

Let w be the weight of fuel burned in a given furnace per second in pounds
V_0 the volume at 32° of the air supplied per pound of fuel.
τ_0 the *absolute* temperature at 32° Fahr., which is 461° + 32°.
τ_1 the absolute temperature of the gas discharged by the chimney, whose sectional area is A; then

Velocity of the current in the chimney in feet per second is

$$= \frac{w \times V_0 \times \tau_1}{A \times \tau_0}.$$

The density of that current in pounds to the cubic foot is very nearly

$$= \frac{\tau_0}{\tau_1}\left(0{\cdot}0807 + \frac{1}{V_0}\right);$$

that is to say, from 0·084 to 0·087 × ($\tau_0 \div \tau_1$).

Let l denote the whole length of the chimney, and of the flue leading to it in feet;

m its "hydraulic mean depth;" that is, its area divided by its perimeter; which, for a square or round flue and chimney, is one quarter of the diameter;

f, a coefficient of friction, whose value for currents of gas moving over sooty surfaces is estimated by Peclet at 0·012;

G, a factor of resistance for a passage of air through the grate, and the layer of fuel above it; whose value, according to the experiments of Peclet on furnaces burning from 20 to 24 pounds of coal per square foot of grate, is 12.

Then according to Peclet's formula,

The *head* required to produce the draught in question is

$$= \frac{\mu^2}{2g}\left(1 + G + \frac{f.l}{m}\right),$$

which, with the values assigned by Peclet to the constants, becomes

$$= \frac{\mu^2}{2g}\left(13 + \frac{0{\cdot}012 \times l}{m}\right).$$

When the *head* is given the value of μ may be calculated, and then,

Weight of fuel which the furnace is capable of burning *completely* per hour

$$= \frac{\mu \times A \times \tau_0}{V_0 \times \tau_1}.$$

It is usual to reckon the *head* by taking 1 inch of water as the unit; then,

$$\text{Head in inches of water} = 0{\cdot}192 \times h \times \frac{\tau_0}{\tau_1}\left(0{\cdot}0807 + \frac{1}{V_0}\right).$$

Mr. Thornycroft found, by careful experiment with steam launches and torpedo boats having "loco" boilers and working with a *plenum* (that is, with a closed stokehole into which air is forced), "that of the initial pressure, the resistance of the tubes accounts for about seven-tenths of the whole, the resistance of the fires and fire-bars being only about one-tenth;" and that "the pressure in the funnel, as measured, was sensibly equal to atmospheric pressure."

Professor Rankine also stated that if H be the height of the funnel, τ_2 the absolute temperature of the external air, then—

Head produced by chimney draught

$$= H\left(0{\cdot}96\frac{\tau_1}{\tau_2} - 1\right).$$

or, taking h as the head,

Height of chimney required to produce a given draught

$$= h \div \left(0{\cdot}96\frac{\tau_1}{\tau_2} - 1\right).$$

The velocity of the gas in the chimney is proportional to $\sqrt{h}$, and therefore to $\sqrt{0{\cdot}96\,\tau_1 - \tau_2}$.

The density of that gas is proportional to $\frac{1}{\tau_1}$.

The weight discharged per second is proportional to velocity $\times$ density, and, therefore, to $\frac{\sqrt{0 \cdot 96\, \tau_1 - \tau_2}}{\tau_1}$; which expression becomes a maximum, when $\tau_1 = \frac{25}{12}\tau_2$. Therefore the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12.

When this condition is fulfilled $h = H$.

That is, the height of the chimney for the best draught is equal to the *head* expressed in hot gas, and the density of the hot gas is half that of the air.

Then, if T_1 be the temperature in degrees Fahr. of the air, the best temperature of chimney base is $= \frac{5,993 + 25\,T_1}{12} = 500 + 2 \cdot 08\,T_1$.

Then the following holds good when $H = h$.

Temperature of external air, - -	40° F.	50° F.	60° F.	70° F.	80° F.	90° F.
Best temperature at funnel base, - -	583°	604°	625°	646°	666°	687°

The Size and Height of Funnel are governed in practice rather by circumstances than by scientific investigation; the diameter is fixed by arbitrary rules based on successful practice, and the height such as suits the appearance or service of the ship. Of late years tall funnels have been the fashion.*

Rule.— Height of funnel $H = \cdot 007\left(\frac{C}{A}\right)^2$;

$$A = \frac{C \times \cdot 084}{\sqrt{H}}.$$

H is height in feet; A, the area of section in square feet; and C, the consumption in pounds per hour of fuel on the grates connected to the chimney.

Evaporation.—The heat of the gases from the furnace should be absorbed by the surfaces with which they come in contact on their passage to the chimney, and the efficiency of this part of the boiler depends on the capability of those surfaces readily to take up the heat, on the material to transmit it by conduction to the inner surface or that with which the water is in contact, and on that inner surface being in such a condition as to give up the heat to the water. The *internal* efficiency of the boiler depends on the convection or circulation of the water in the boiler, whereby fresh portions are successively brought in contact with the hot surfaces. The importance of this latter factor was seldom sufficiently appreciated in estimating the efficiency of a boiler, although now engineers give it first consideration.

When the furnace is internal—that is, when it forms a part of the boiler proper and is surrounded with water—a large proportion of the total heat of combustion is absorbed by it, partly by direct contact with the hot fuel at the sides, and partly by radiation from the glowing surface of the incandescent fuel when coked. The furnace also absorbs heat from the hot gases passing along its surface.

* The funnels in Atlantic express steamers are now 160 to 180 feet high above the grate levels, oval in cross-section and as large as 24 feet $\times$ 17 feet.

The furnace or fire-box of a locomotive boiler is usually made of copper, and it has been proved by experiment that a very large proportion of the whole heat generated in it is absorbed by it, consequently its evaporative efficiency is very high. The furnace of the ordinary marine boiler is of steel, which is not so good a conductor of heat as copper, and, therefore, does not transmit such a large proportion of the total heat of combustion, although its evaporative power is still high. (For conductivity of metals see Chap. xxx.)

Combustion.—The combustion of the fuel is not always *completed* in the furnace of the marine boiler, as the gases distilled from it during the process of coking escape to the chamber beyond the furnace before sufficient air has been supplied to complete combustion; and also it sometimes happens just after firing that the temperature *above the fuel* is not sufficiently high to cause ignition. Moreover, the carbon often unites in the furnace with only sufficient oxygen to form carbon monoxide, and this flows unchanged into the combustion chamber; if, then, a further supply of air is found there, another portion of oxygen is taken up, and carbon dioxide or carbonic acid gas is formed; that is what usually takes place with bituminous coal, and unless this second supply of warm air is provided, the combustion of a large portion of the fuel is not completed in the boiler. If the fire is completely covered with *green*—that is, fresh fuel—the part next the hot surfaces gives up its volatile elements, which rise and become cooled in expanding, so that when they come in contact with the oxygen of the air, the temperature is not high enough to cause them to chemically combine—that is, to ignite, and consequently they merely mix mechanically and flow on until they pass out at the mouth of the funnel and appear as smoke.

If the smoke-boxes leak air, as they only too often do, ignition will sometimes take place in them, and flame will ascend the funnel and appear at the top, where there is a further supply of air sufficient to complete combustion.

When, however, by careful stoking and regulating the supply of air, so that combustion is completed in the combustion chamber, the evaporative power of that part of the boiler is high, and with the furnaces is the most valuable part of the boiler for transmitting heat to the water.

Heating Surface.—As has been stated, the efficiency of the heating surface of the boiler depends on the material, its thickness, and on the state of the surfaces in contact with the hot gases and water. The furnaces and combustion chambers are of steel, whose conductivity is inferior to that of copper (16 to 91), but is still good.

The superior evaporative power of the furnace is due in great measure to the cleanness of the surface exposed to heat; there is no deposit of soot or ash on it, and the smallest possible amount of oxide; the combustion chamber also is generally in the same condition. The roughness of the surface exposed probably increases its power of receiving heat, not so much from any abstract virtue in that state, as from the *actual surface* being greater than if smooth. Very much depends on the condition of the inside surface exposed to the water; if it is quite clean and absolutely *smooth*, it is not so efficient as slightly dirty and rough; the best condition being roughness with freedom from coating of bad conductors. If a metallic surface is smooth and clean, evaporation from it is slow and intermittent, because on it is formed a film of steam, which is a bad conductor, and this will only disperse when its buoyancy overcomes the attraction; when the attraction is overcome, the steam rises suddenly *en masse*, the surrounding water then flows into its place, when a film is again formed, and a bubble accumulates as before. If, on the other hand, the surface be rough, the film is broken up by numerous points on the surface, which serve as accumulators and starting points for myriads of small bubbles; these are formed quickly, rise freely and continuously, giving a rapid and steady supply of steam. It is for this reason that many boilers *prime*—that is, work with violent ebullition—when quite new

Tube Surface.—The tubes of a marine boiler represent the greater portion of the total heating surface, and it is to this part that great attention is required, both in designing and working a boiler. They are usually made of iron in the mercantile marine, and of steel in naval boilers. The conductivity of the brass, which was formerly used in the Navy, is theoretically double that of the iron (30 to 16); but in practice with foul surfaces, there is not really so great a difference, their relative value being then about 3 to 2; to obtain, therefore, results commensurate with their extra cost, brass tubes had to be kept very clean always.

Much stress is sometimes laid on certain experiments which show the relative evaporative power of certain portions of the tubes, to prove thereby that the final foot, or two feet, of tube is practically useless in *every* boiler. It is not very surprising to find that the first foot of tube has a high evaporative power, and that the power decreases rapidly the farther removed the portion is from the furnace or combustion chamber. Considering that the temperature of the gas on entering the tube is 2,000° to 2,400°, and on leaving it should be not more than 600°, while the temperature of the water surrounding the tube is very nearly the same (386° F.) at every part, the transmission of heat should be very much greater at the entering end, where the difference is 1,700° to 2,000°, than at the other, where it is only 220°; moreover, the end next the funnel is more liable to get dirty from deposit of soot, etc., and so to rapidly fall off in efficiency. But in continuous practice it was not found that there was such a large difference in the value of the two ends of the tubes, because the end at which there is the rapid evaporation soon became covered with scale, and its efficiency reduced when salt feed water was used; the gas temperature was then not so much reduced after passing through the first portion, and for this reason the last portion was found to have a higher efficiency than before. The size of tube has also some influence on its efficiency, for whereas the surface increases as the diameter, the contents increase as the square of the diameter. If a tube, then, of 4 inches diameter be substituted for two of 2 inches diameter, the surface is the same, while the cubic capacity is doubled; if the rate of flow be the same in both cases, the 4-inch tube will pass twice the quantity of hot gas through it that flows through the two 2-inch ones, and have only the same surface to absorb and transmit the heat. If the quantity flowing through the tubes be the same in both cases, the 4-inch tube is still at a disadvantage, inasmuch as the mean distance of the gas from its surface is greater than that in the 2-inch ones. The velocity through the small tubes will, in the latter case, be double that in the 4-inch one, and will therefore cause a brisker circulation of the hot gas, and the liability of soot and ash deposit is considerably reduced. Hence tubes of smaller diameter are used with advantage with forced draught, for the same evaporation is effected with smaller amounts of surface. The "Serve" tube made by J. Brown & Co., having 6 or 8 ribs projecting inside, has the great advantage of large absorbent surface in proportion to its cubic capacity. The twisted strips of iron placed inside the tubes by Mr. Howden, and called "retarders," have the effect of reducing the cubic capacity as well as circulating the hot gases; the retarders, however, can transmit very little of the heat they absorb, in the way the ribs of the Serve tubes do.

Evaporative Power.—The probable evaporative power of a boiler may be found approximately by the following formula:—

Let e_1 be the *theoretical* evaporative power of the fuel, F the weight of coal burned on the grate in pounds per hour, and K the total heating surface in square feet; then

Pounds of water evaporated per pound of fuel burnt

$$= 1{\cdot}833 \left(\frac{K}{2\,K + F}\right) e_1.$$

Example.—To find the evaporative power of a boiler which burns a fuel whose theoretical evaporative power is 15, the number of pounds burnt per hour on the grate is 800, and the total heating surface is 1,000.

Here $$E = 1{\cdot}833 \left(\frac{1{,}000}{2{,}000 + 800}\right) 15, \text{ or } 9{\cdot}825 \text{ lbs.}$$

The efficiency of the boiler is, by this rule,

$$1{\cdot}833 \left(\frac{K}{2\,K + F}\right), \text{or } 0{\cdot}655.$$

This rule is only approximate, however, because the heating surface is not always wholly *real* heating surface, some being only nominal; many boilers as made formerly have given better results after the removal of tubes, which very materially decreased the nominal heating surface. If a boiler has just enough surface to absorb heat from the gases, so that the temperature at the funnel base is only such as is sufficient to produce the required draught, then the heating surface is effective; any surface added to this is superfluous, and in very many cases does positive harm.

The Efficiency of a Boiler under various conditions of working can be ascertained as follows:—

W is the weight of water actually evaporated per pound of fuel.
C ,, calorific value of the fuel used in British thermal units.
H ,, total heat absorbed in raising a pound of steam from a pound of water at 32° F., and h that of the feed-water.
T ,, temperature of the steam produced in degrees Fahr.

If superheated steam is produced at a temperature T_s, so that the amount of superheat is $T_s - T$, the additional H_s will be found by multiplying $(T_s - T)$ by the specific heat of steam.

(1) Efficiency of the boiler $= \dfrac{W(H - h)}{C} \times 100$ per cent.

(2) Efficiency of superheating $= \dfrac{W(H + H_s - h)}{C} \times 100$ per cent.

The factor of evaporation $= \dfrac{(H + H_s - h)}{966}$.

The specific heat of steam at, say, 200 lbs. absolute and superheated to 480° F. is 0·59, and if to 600° F. it is 0·545. This is the mean specific heat between 381° and 480° and 600° F., and is the ratio of it to the heat required to raise a pound of water by the same amount. Taking the British thermal unit as 778 foot-lbs., it requires on the average 459 foot-lbs. per degree to superheat a pound of steam of 200 lbs. pressure to 480° F.; or, in all, $(480 - 381) \times 0·59$, or 58·3 B.T.U.

The temperature at uptake depends on the temperature of the water in the boiler; for instance, at 300 lbs. pressure, it will be 422° F.; so it will be no use to try to bring the temperature of the gases below 400° or even 450°, and with such a temperature there the efficiency of the tubes will be very low at 200 lbs., inasmuch as the water then is 387° F. It is, therefore, manifest that it is better to have a higher temperature at the smoke-box, and reduce at funnel base by air-heaters, feed-heaters, or super-heaters to dry the steam.

CHAPTER XXII.

BOILERS—TANK BOILER DESIGN AND DETAILS.

There are in use on shipboard to-day two distinct types of boiler, viz. :—

(1) **The Tank Boiler,** consisting of an external shell, in which is contained the water and the means whereby that water may be converted into steam; it also has within it a space for the temporary storage of the steam when formed; and

(2) **The Water-Tube Boiler,** consisting of a structure of tubes joined together, intercommunicating, and connected to a receiver at the top, and generally to water chamber or chambers at the bottom. The water is in this class always within the tubes, and circulates when heat is applied to their surfaces. The fire is generally on a grate immediately below the nests of tubes, so that the hot gases ascend and pass through them.

The Tank Boiler to-day exists in various forms, but the shell is always cylindrical or nearly so. The furnaces within it are always cylindrical in general form, but they are often stiffened to resist collapse by corrugations or equivalent means. The furnaces deliver the hot gases into a chamber, within which combustion may be completed, and the contents distributed over a large number of small tubes, through which they are conveyed to the smoke-box, and by which their heat is absorbed on the way there, so that they play the part of a surface condenser to them. They pass the heat into the water, whereby it is heated and some of it converted into steam. In the past there have been many forms of tank boiler other than the cylindrical one, of which

(*a*) **The Oval Boiler,** as shown in fig. 204, which, when side room is limited. permits of a larger and more powerful boiler than if cylindrical; and although not so *light* per unit of power as a cylindrical, it preserves most of the good features of that form, together with a simple system of staying whereby strength is obtained.

(*b*) **The Vertical or Haystack Boiler**, having a cylindrical shell with a hemispherical top, as in fig. 213. Or it may have a flattened or oblate spherical top with the funnel passing through it, as in fig. 212.

(*c*) **The Locomotive Boiler** in a modified form was used very generally for small craft, and even for destroyers of 3,500 I.H.P., but they were not altogether successful, especially under severely forced draught. The form of locomotive boiler, as shown in fig. 214, was, however, quite satisfactory in torpedo gunboats of considerable power with a fairly high air pressure.

The Boiler used in the Mercantile Marine is to-day almost exclusively of the tank type, and simply cylindrical in form, made of steel, and fitted with iron or steel lap-welded tubes. The furnaces are often, in small ships, plain cylinders welded and flanged to meet the combustion chamber in such a way that they can be passed through the hole, provided for them in the front plate of the boiler (*v.* fig. 206). In larger and more important ships, where supervision and care is better observed, the furnaces are of the corrugated or ribbed variety.

Cylindrical Boilers with Two Furnaces, as in fig. 205, are used in small

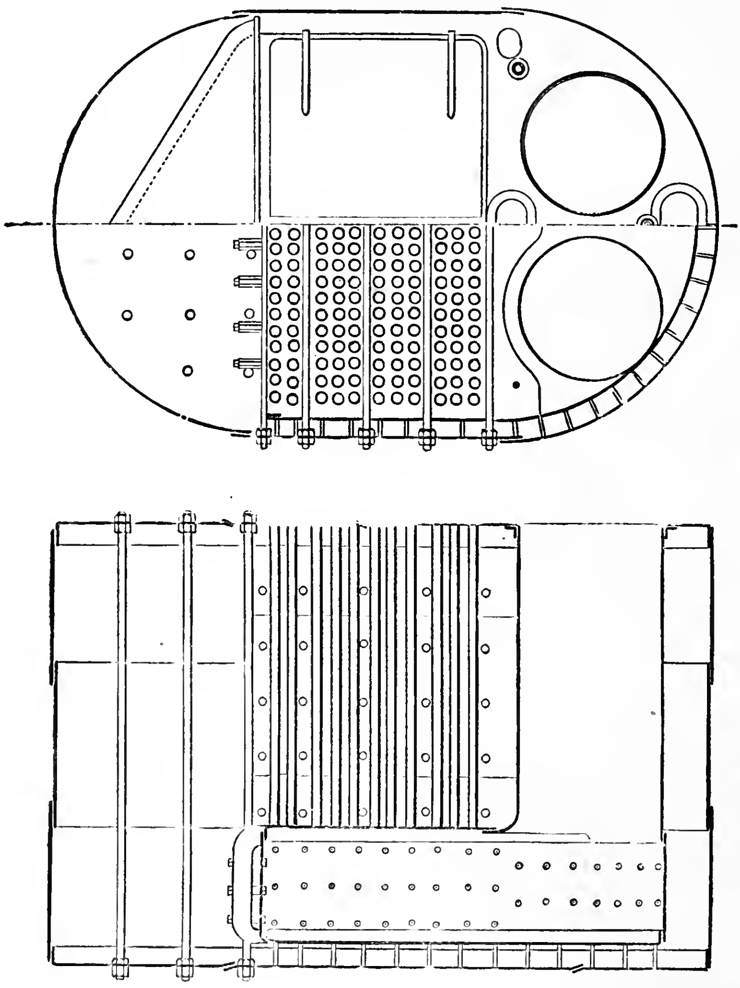


Fig. 204.—Oval Boiler with Two Furnaces and One Combustion Chamber.

ships where one such is sufficient, and in other shallow ships having more than one where it is desired to have the boilers stowed under decks, as in yachts and some passenger steamers; more than one boiler is often decided on as a means to ensure immunity from disablement in case of derangement of one of them. Two furnace boilers may be made of considerable diameter if required, as the furnaces as now made may be large—as much as 54 inches in diameter for moderate pressure, and 50 inches for 200 lbs. per square inch.

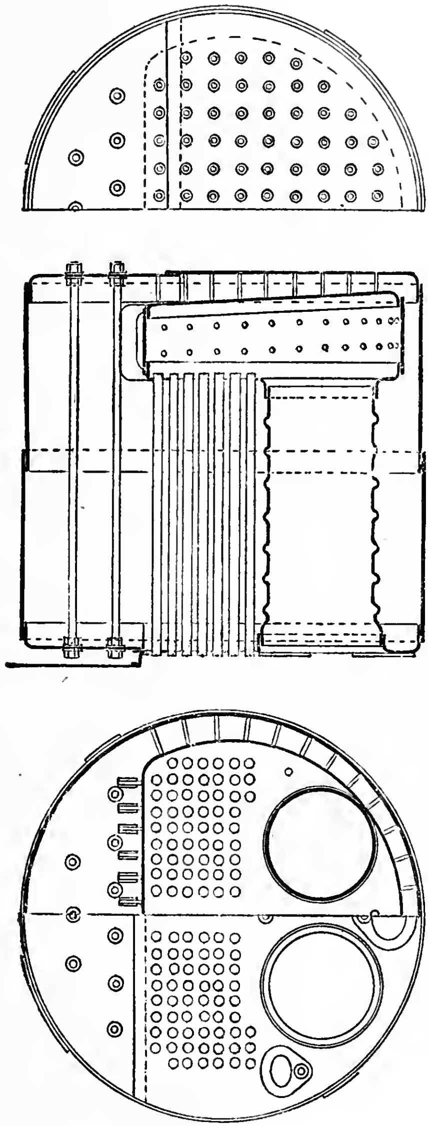


Fig. 205.—Single-ended Cylindrical Boiler with Two Furnaces and One Combustion Chamber.

Cylindrical Boilers with Three Furnaces are in very general use, and are made up to diameters of 15·8 feet. It is usual now to fit such boilers with a separate and distinct combustion chamber to each furnace, as shown in

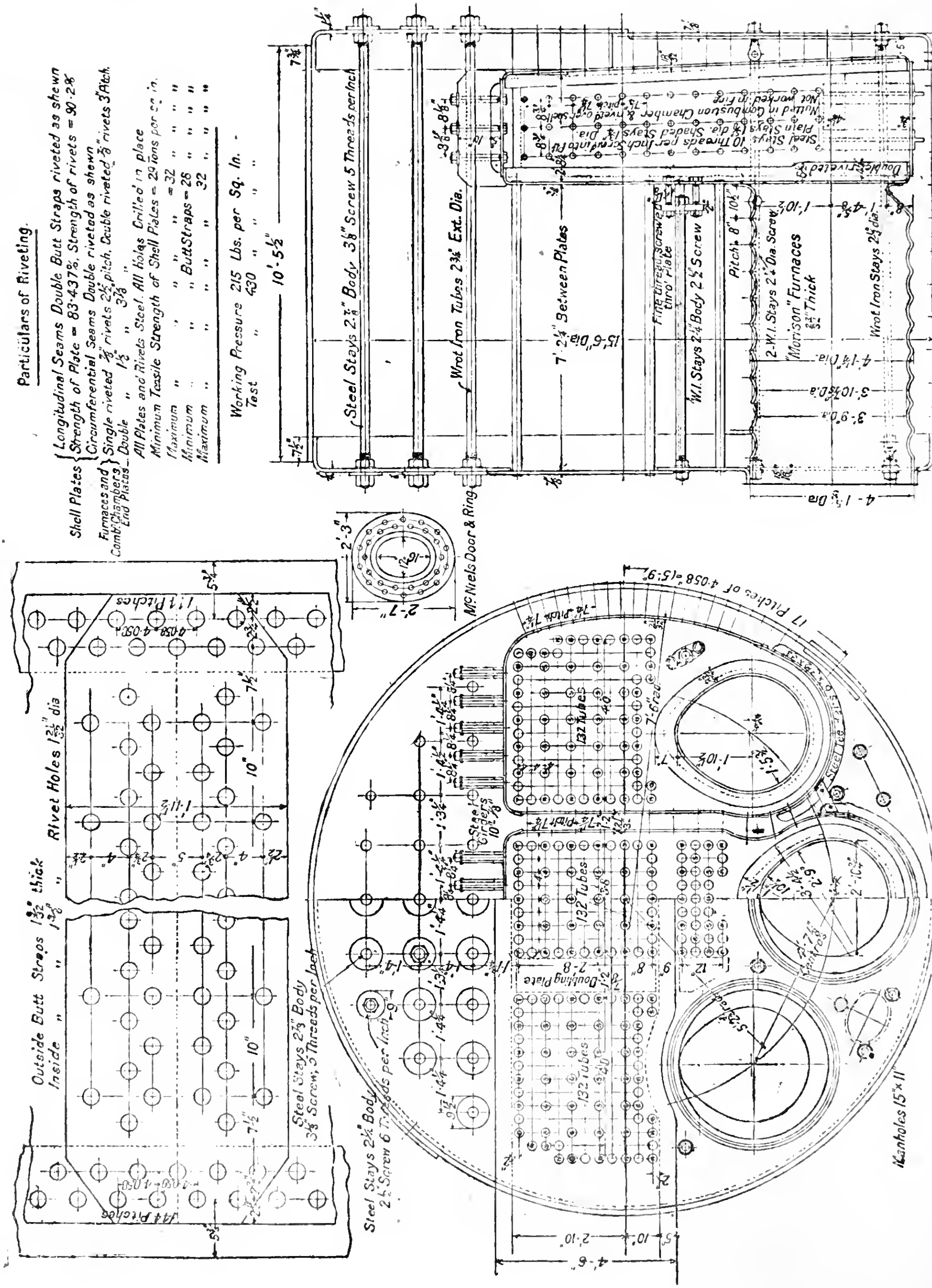


Fig. 206.—Three-Furnace Single-ended Boiler with Three Combustion Chambers.

fig. 206, end view; but it has been made often in the past with one chamber common to the three.

Cylindrical Boilers with Four Furnaces may be as shown in fig. 207, with the furnaces connected in pairs to each of two combustion chambers, or

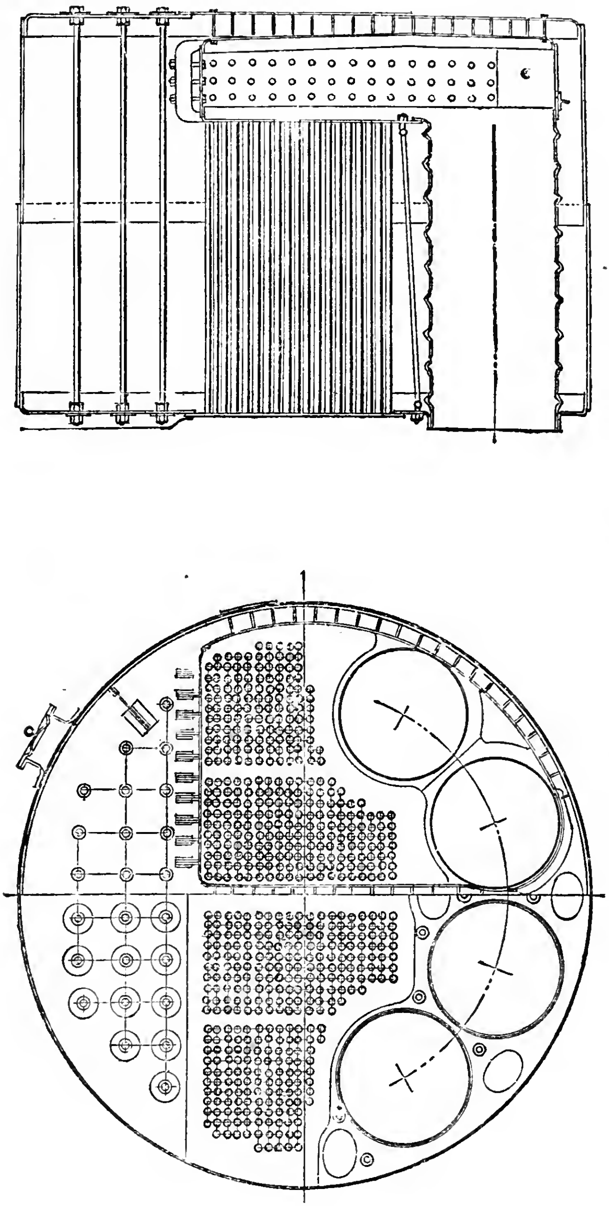


Fig. 207.—Four-Furnace Cylindrical Boiler with Two Combustion Chambers.

with the two middle furnaces connected to one common chamber, and each wing furnace with its own chamber. This boiler may, and often is, made

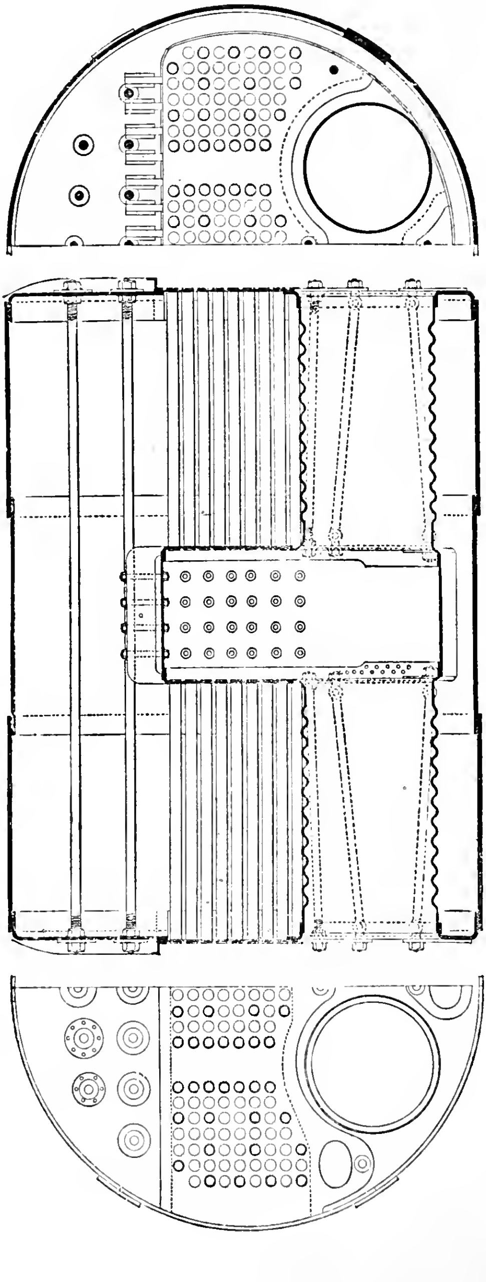


Fig. 208.—Double-ended Boiler with One Combustion Chamber.

with a chamber to each furnace; or, as is seldom done now, all four furnaces may have one chamber common to them all.

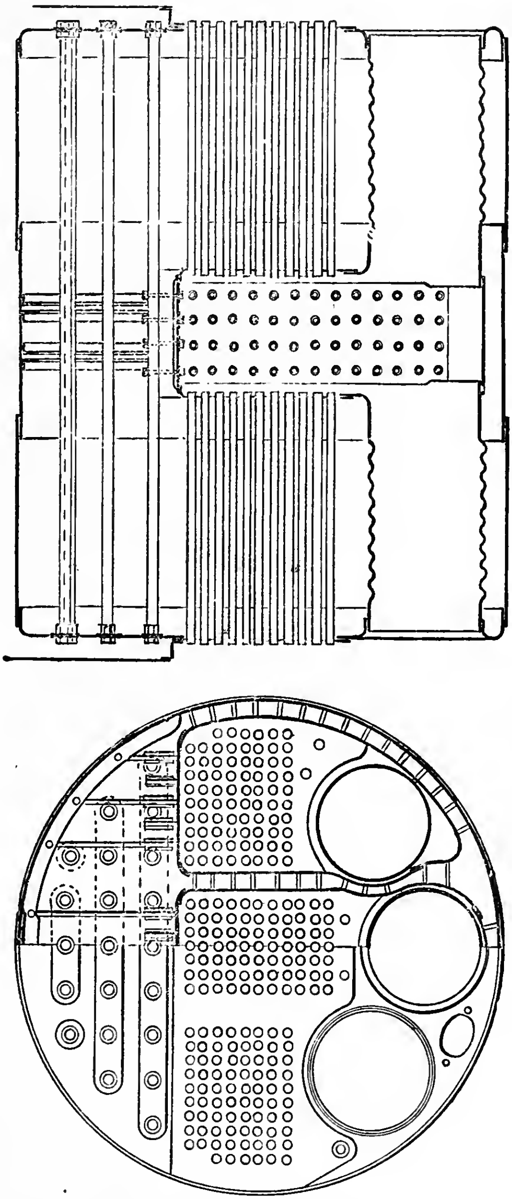


Fig. 209.—Double-ended Boiler (Fox's Furnaces).

The Double-ended Boiler has furnaces at each end, each with its own chamber, or opposite furnaces having a chamber common to them, as in

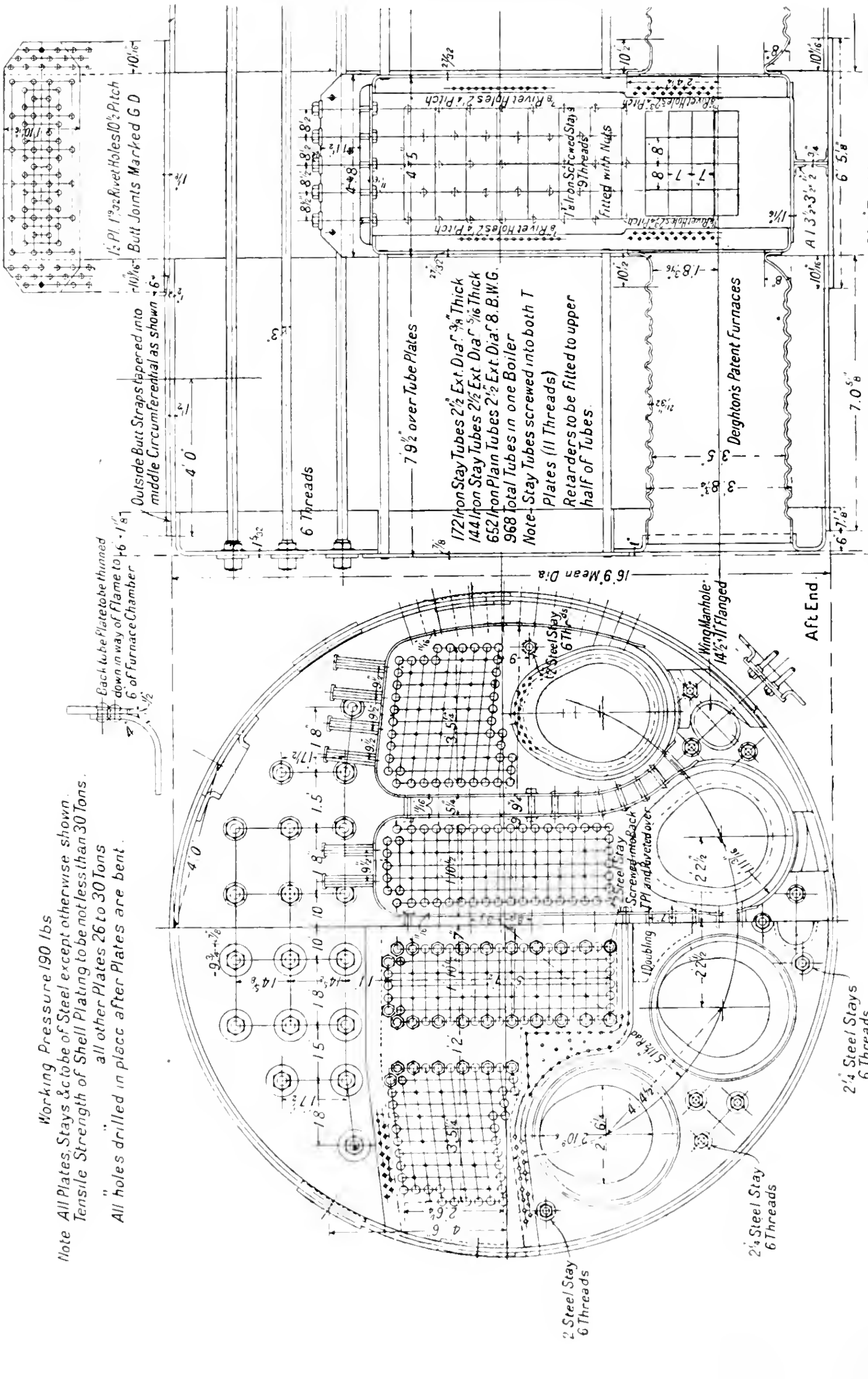


Fig. 210.—An Eight-Furnace, Four Combustion Chamber, Double-ended Boiler.

fig. 209; or there may be, and often was, to save weight and space, one chamber common to four furnaces, as shown in fig. 208. Double-ended boilers are now made of enormous size, as much as 18 feet diameter and as large as 24 feet long.

The double-ended boiler is lighter and cheaper to make per unit of power than a single-ended, and occupies less space. Being of smaller diameter than a single-ended one of equal power or heating surface, the shell can be made of thinner plates, which for high pressures used to be a matter of serious consideration; but to-day, when steel makers can and do roll plates of any thickness up to 2 inches of large area at moderate costs, and most boiler-shops are equipped with tools capable of handling and dealing effectively with such plates, this arrangement has little or no weight. But the straining effect of high temperature, differences of temperature, and changes of temperature are as severe and active now as they ever were; and although the care and methods of the modern boiler-maker are beyond all praise, and produce work far superior to that of a few years ago, the effects of racking are liable to appear, although they are not so evident in the form of leaks as were only too often the case formerly; it still remains, therefore, that the single-ended boiler can be trusted to withstand rougher usage than the double, and is, therefore, more generally used in the mercantile marine than formerly. Moreover, since quite large units of the single-ended type are made now, there is not the same necessity for the double-ended in full-powered express steamers as obtained formerly; nevertheless, in these latter the double-ended boiler is still frequently used to economise space and weight. In the Navy the double-ended boiler was never a favourite with the engineers responsible for them, and although largely used in cruisers on account of their lighter weight from 1887 to 1897, they have since been seldom used, and latterly the tank boiler has been abandoned altogether. There was, however, a special form of cylindrical boiler introduced into the Navy so far back as 1854, which has survived, and has been often adopted for use in express steamers, and known as

The Gunboat Boiler (fig. 211), with the furnaces at one end and the tubes at the other, with the combustion chamber between them, is a very efficient boiler, and the lightest cylindrical one per unit of power. It is a very convenient design for shallow ships having plenty of fore and aft room to stow the boiler in.

The Naval Boilers of to-day are, however, almost exclusively single-ended cylindrical and water-tube boilers of sorts; in small ships exclusively the latter. In larger ships the rule was to have a portion of water tube with a portion of cylindrical, the latter being used for cruising at slow speeds, and the former added to them when high power or full speed is required; now water-tube boilers only are used.

In the Mercantile Marine the Water-tube Boiler has practically found no favour in this country, and not very much elsewhere; it has undoubtedly some good claims for serious and favourable consideration, especially for certain services, and it is very remarkable that on shore the very large installations at electric generating stations water-tube boilers are almost exclusively used, in spite of the fact that electrical engineers are very keen observers of fuel consumption, and that they have had experience with quite good forms of the cylindrical boiler at some of these stations. It is, of course, admitted that in the ordinary merchant steamer on long voyages there is for days no variation in the demand for steam, and from noon to noon hardly a half revolution difference in speed, whereas at an electric light station the demand for steam is always changing, and a sudden fog may make a sudden demand for an enormous increase in it, which necessitates the forcing

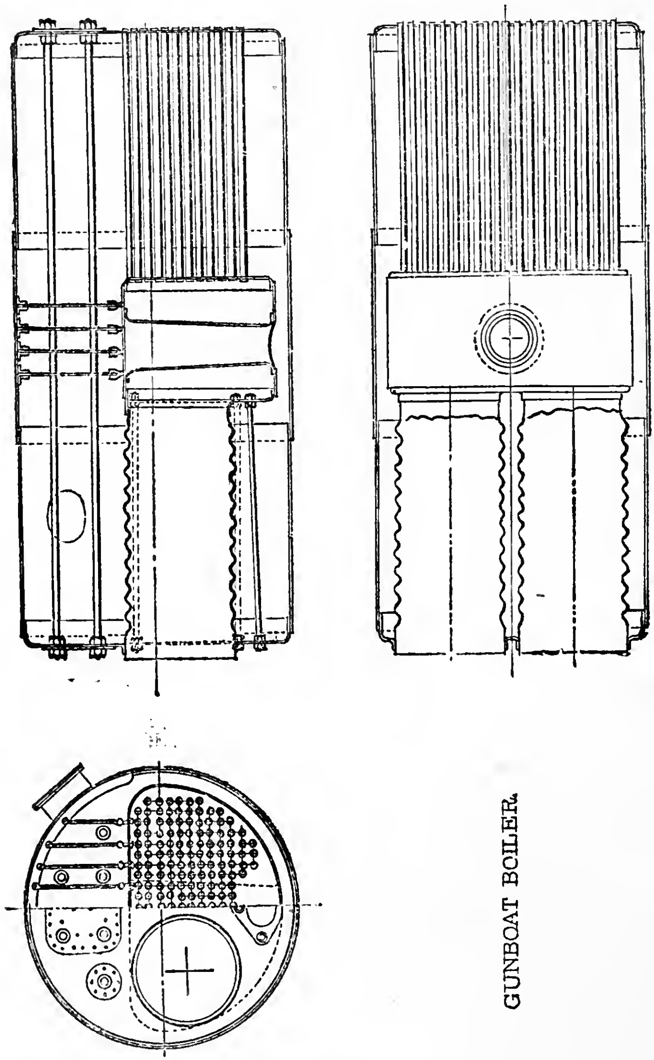


Fig. 211.—Naval Boiler, Gunboat Type.

of such boilers as are on service, as well as the addition to their number as rapidly as they can be possibly got into action; in both cases the boilers are severely tried, and only those which can withstand the racking due to sudden changes of temperature can be employed.

Small Boilers have usually two furnaces, and with this number are more efficient than with three even when of moderate size.

The number and size of the furnaces must, however, depend on the size of the boiler and the heating surface it is to contain. It is found in practice that large furnaces are more efficient as coal consumers than small ones, and the reason is not far to seek. The grate area with the same length of fire-bar increases as the diameter, while the section through which the air passes, both above and below the bars, increases as the square of the diameter; it is also possible to give a good inclination or rake to the bars with a large furnace, which very materially assists combustion. In practice the grates are not of course always of the same length, but they do not increase in length as the furnace does in diameter, and consequently the air passages increase more rapidly than does the grate area when the diameter of furnace is increased. Furnaces should be not less than 36 inches, nor more than 48 inches in diameter, except under special circumstances. Taking this as a rule for guidance, boilers may be made up to 9 feet diameter with one furnace, up to 13 feet 6 inches diameter with two furnaces, up to 15 feet with three furnaces, and beyond that diameter four furnaces are necessary to avoid too long length of grate.

A single-furnace boiler has of course one combustion chamber, a two-furnace boiler may have one chamber common to the two furnaces (fig. 205), or a separate one to each. When there is only one boiler in the ship the latter plan is preferable, as then the bursting of a tube cannot wholly disable the boiler; when there are two or more boilers one chamber common to the two furnaces is preferable, as by stoking the fires alternately an even supply of steam is kept up and the smoke consumed. A three-furnace boiler has usually three separate combustion chambers, and the same remark applies to it as to the two-furnace boiler. The four-furnace boiler has generally only two combustion chambers, one wing and one middle furnace having a common chamber; but some engineers prefer three chambers, the two middle furnaces having one in common, and each wing furnace a separate one.

The chief objection to two large furnaces instead of three smaller, and to three larger ones instead of four smaller, is the longer grate required to get the requisite area, and to the large amount of dead water between the furnaces at the bottom. There is also to be considered the limit placed by the Board of Trade and Registry rules to avoid risk of collapsing by direct crushing of the metal, which prevents the adoption of the larger furnace with very high pressures.

It is unusual and certainly most difficult to use plates above $1\frac{3}{4}$ inches thick in the construction of a boiler shell, and it is this consideration which often fixes the limit of diameter. Boiler-shop appliances are now made to deal with even thicker plates when necessary. Boiler shell plates up to 2 inches thick can be obtained 12 feet wide and of 250 square feet area at quite moderate prices in mild steel from 27 to 36 tons ultimate tensile strength.

The Double-ended Boiler (fig. 209) has furnaces at both ends with return tubes over them, and is generally tantamount to two single-ended boilers back to back, but with the backs removed. It is made up to 18 feet diameter and as much as 24 feet long; but such very large boilers are unusual, partly owing to the want of facilities for moving such great weight, and partly because the conditions under which such large boilers are possible are limited to very large steamers. For the lower pressure of steam for turbines such large boilers are now frequently supplied.

The double-ended boiler is lighter (*vide* Table lii.) and cheaper in proportion to the total heating surface than the single-ended boiler, and its evaporative efficiency in practice is generally higher. On the other hand, greater care is necessary in designing and in working it. That it is lighter is obvious, and that it is cheaper may be inferred from the fact that there is less material, and less labour consequent on the reduced quantity of material.

The simplest form of this kind of boiler is one in which all the furnaces open into one common combustion chamber (fig. 208); this form, although at one time common enough, is now seldom adopted. The objections to it are, that the bursting of one tube may disable the whole, that the cleaning of one fire causes the efficiency to sink very low on account of the whole being effected by the inrush of cold air, and that unless special means be provided to promote proper circulation there is a strong tendency to *prime*. Some of these objections are got over by building a thin brick wall with special bricks across the middle. Mr. Howden recommended this arrangement with his system of forced draught, such being necessary whenever there is a chamber common to opposite furnaces.

The next simplest form is one in which opposite furnaces have a combustion chamber in common (fig. 209)—that is, it differs from the first by having the combustion chamber divided longitudinally by water spaces. This avoids the chief objections raised against the first form while retaining its chief advantages, which are, simplicity of construction, by discarding the flat back of the combustion chambers, with the necessary stays, etc., and the greatest heating surface within the smallest limits of length. It is often urged against this form of boiler that the tubes are very liable to leakage at their back ends, arising from the rush of cold air against the tube plates when the door of the furnace opposite it is open, causing it to buckle. It sometimes happens that the tubes in this kind of boiler do show a tendency to leak, but it is then generally due to the want of expansion on the part of the first row of stays above the combustion chamber when they are placed too close to the tubes. If these stays are at least 12 inches above the tubes, so as not to hold the front tube plates too rigidly, then when steam is being got up the expansion of the tubes simply causes the plates to spring very slightly instead of to start their ends and cause them to leak. The leakage from springing of the tube plate from exposure to cold air can only take place when the combustion chamber is unduly short, and when there is an insufficient number of stays to the tube plates.

This particular form of boiler is very generally used for large power and high speed; the evaporative results obtained from it are most satisfactory. Care, however, is required in raising steam, and the opening of fire-doors to check evaporation is a reprehensible practice at all times for them, as it is indeed for all boilers. A brick semi-partition in the middle of the combustion chamber will prevent the cold air rushing on to the opposite tube plate, and it acts also as an equaliser of temperature in the combustion chamber at all times. If, however, the combustion chamber is too small, this will only magnify the defect by causing intense local heat, and thus tend to crack the plates.

Another common form of double-ended boiler has the furnaces at one end with one chamber common to them, and those at the other end with another chamber in common. The boiler is then longer than either of the other forms, and more expensive; the combustion chambers have large flat backs, requiring a very large number of stays, which prevent their being properly cleaned from scale.

The last form, which is by far the most expensive and heaviest, but now often adopted, is one in which each furnace has an independent combustion chamber. There is little need of description, as it is to all intents and purposes two single-ended boilers, except that the water and steam are common to the two parts.*

Oval Boilers are included under the generic term of cylindrical, as they partake of the principal features of that class. The transverse section is, however, not an ellipse, but is really formed by two semicircles with a rectangle intervening. The flat sides thus left between the semi-cylinders require staying, the first rows being at the commencement of the flat. There are both single- and double-ended oval boilers, which for pressures under 120 lbs. may be made both simply and economically to very large sizes, as the thickness of shell plate depends on the diameter of the cylindrical part. Two very large furnaces may be thus fitted into a cylindrical part of comparatively small diameter, sufficient heating surface being obtained by giving the requisite height. This form is most convenient when the boilers have to be stowed fore and aft, and the diameter is limited by the breadth of the ship between the stringers; also when forced draught is employed they permit of the large amount of heating surface in proportion to the grate area which is necessary for economic results.

Holt's Boiler.—An ingenious form of double-ended boiler, used formerly by the late Alfred Holt, consists of two oval end parts united by a cylindrical part whose axis passes through the upper focus of the oval; in the bottom of the end parts are the furnaces, which are each connected by a large tube to a combustion chamber in the middle of the cylindrical part, from which the tubes extend to the front above the furnaces, and are consequently much longer than usually found in a marine boiler. These boilers are made of great length, and practice proved them to be very good and efficient steam generators, lasting very much longer than the ordinary double-ended boiler, being more elastic and the middle portion raised far above the influence of bilge-water.

Dry Combustion Chamber Boiler.—The boiler in this case is a simple cylindrical or oval shell, having the furnaces extending from end to end, and the tubes over them likewise from end to end. The combustion chamber is external, and does not form an integral part of the boiler; but is built of brickwork so as to enclose one end, and form a connection between the furnaces and tubes. If two such boilers are placed back to back and some distance apart, and the space between them enclosed by brickwork, a double-ended arrangement is formed similar to the first described, except that the water and steam are not in common to the two parts, except by means of special connections. Such a boiler is very cheap to manufacture, as the wet combustion chamber with its flat sides and stays is avoided; the advantage of two boilers without their cost is obtained, and the evaporative

* The naval practice is to have a separate chamber to each furnace in all boilers.

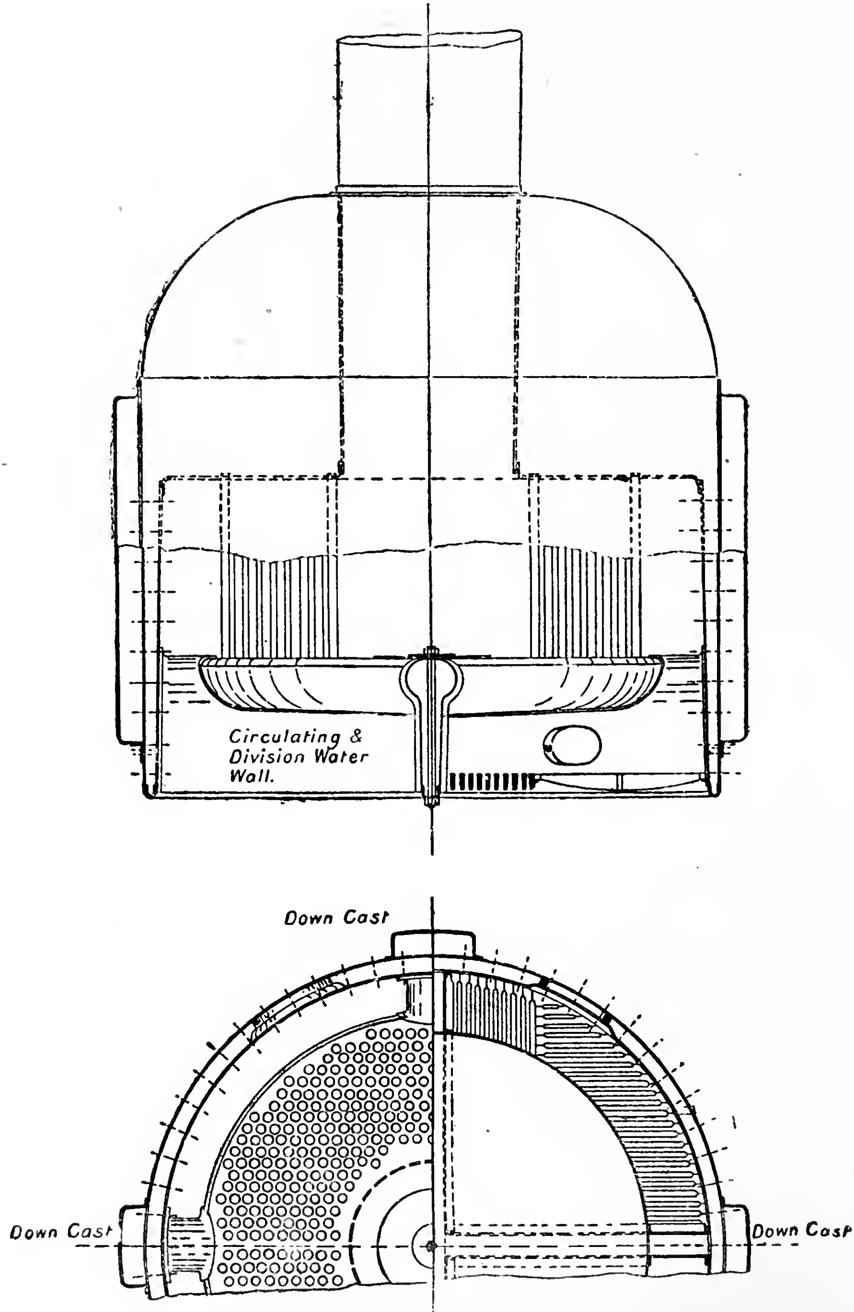


Fig. 212.—Scotch Haystack Boiler.

efficiency is very high. The intense heat from the brickwork, however, is liable to crack the tube plates, unless care is taken; but if ample space is provided in the combustion chamber, there is not so much danger of this, or of damaging the tube ends.

Such boilers are now, however, not so much in use; although, among other things, they accomplish a satisfactory consumption of smoke, and a very even rate of evaporation. Their reputation, however, has been on some occasions somewhat damaged by a too intemperate advocacy, and by improper construction and design of the combustion chamber when attempting too much with them.

For pressures above 100 pounds in ships liable to rough usage, such boilers may be adopted with advantage on account of the absence of the internal combustion chamber.

Gunboat Boilers (fig. 211).—This type of boiler, first adopted by the Admiralty for gunboats, and afterwards chosen for corvettes, is something between the locomotive and cylindrical boiler. The shell is cylindrical, and contains the furnaces at one end and the tubes at the other, the combustion chamber being in the middle between them; the top of the furnaces is, therefore, level with the top of the tubes, and no part of the heating surface is far removed from the water level. The flame and hot gases flow from the furnaces into the combustion chamber, are there sometimes slightly diverted and spread by means of a hanging bridge, and flow onward with only this slight interruption into and through the tubes to the smoke-box. It is not surprising, then, that this boiler burns its coal freely, and evaporates very quickly and efficiently. Two such boilers, having a total grate area of 68 square feet, and a total heating surface of only 2,200 square feet, supplied steam to triple-compound engines developing over 1,340 I.H.P. The coal consumed was about 36 pounds per square foot of grate per hour; the weight of water evaporated by one pound of coal, at a somewhat reduced speed, was about 9½ pounds.

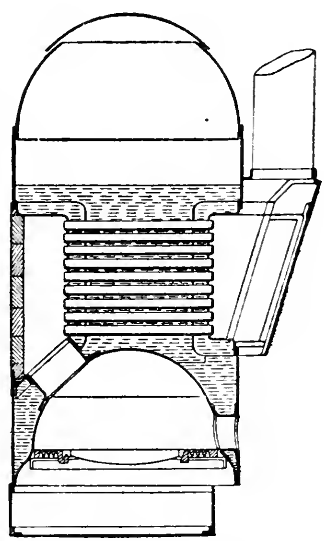


Fig. 213.—Cochran's Boiler.

The chief objection to this class of boiler, which prevents its more general adoption in the mercantile marine, is the great length required; a space is necessary at the back end to get at the tubes, and to admit of the smoke-box, etc.; and the total heating surface is also very small for the space occupied by it. It is, however, very convenient for shallow ships, when it is necessary to have clear decks, and has consequently been used for modern express river steamers, as well as gunboats.

Vertical Cylindrical Boiler.—This kind of boiler, with many variations of internal arrangement, is used for auxiliary purposes, and then generally called the *donkey* boiler. On a large scale it was much used by Scotch engineers for river steamers, where rapid evaporation is of more importance than economy of fuel. It is light and inexpensive in proportion to the grate and heating surface, and occupies a small amount of floor space, capacity being obtained by the height. Fig. 212 is a modern example of such a boiler as has been made for large power. Fig. 213 is a design of vertical cylindrical boiler as invented by Mr. Cochran, now of Annan, N.B., and made by him in large numbers for the auxiliary service of large ships and the main boiler of small ones; it is a very convenient form for these purposes.

Locomotive Boiler.—This boiler has a fire-box of rectangular section, both horizontally and vertically, enclosed in a shell of somewhat similar shape, except that the top is often semi-cylindrical. The tubes are contained in a cylindrical barrel, extending horizontally from the fire-box shell, and at the end of this is the smoke-box. As the name implies, it is similar to the boiler of the ordinary locomotive.

This form of boiler is only employed in the steam launches of the mercantile marine, but was used in the Navy very extensively for torpedo boats as well as launches, and was even adopted on a large scale in torpedo gunboats, as well as in some of the early destroyers. It is a very convenient form for the naval service, as it is, with the exception of water-tube boilers, the lightest kind of boiler for the heating surface contained; and as it is invariably used now with an artificial draught, the smallness of the grate area is no detriment to it. It is also especially well adapted for high pressures, as the flat surface can be stayed without affecting the accessibility, and the cylindrical barrel is of such small diameter, as to be made of very light plates for even very high pressures. Much difficulty is, however, found in keeping the tube ends tight, especially when the draught is forced much; and when pressed to their utmost capability with engines having large cylinders, they are very liable to prime excessively. The only part which presents any difficulty of construction is the furnace crown, which, being flat, requires an extensive amount of stays, etc., and as the evaporation is very rapid from it, there is great liability of a heavy scale being formed if any sea or "hard" water is used, which prevents the heat from passing to the water, and causes it to destroy the plates.

It has the disadvantages of the gunboat boiler, and on that account is not likely to be used in the merchant service, except in very light-draught river steamers in the colonies and foreign countries, where it has for many years done good service.

Wet-Bottom Locomotive Boilers.—The success of the locomotive boiler for naval small craft caused the authorities to adopt them for a bigger class of ship, as well as for a large number of torpedo gunboats. A modification of the design, however, was made, which converted it into a wet-bottom boiler, the air being admitted beneath the bars through an aperture in the front, as in the furnaces of the ordinary boiler. The change, however, was a most unfortunate one, as, with the exception of the design as fitted by Laird Brothers in H.M.S. "Rattlesnake," this form of locomotive boiler proved troublesome and inefficient. The circulation is bad, so that the tube-ends at the fire-box are so seriously affected as to cause considerable leaks and general trouble. The tubes are much nearer to the fire than in the ordinary locomotive boiler, and, consequently, become more readily choked;

in fact, after some hours' steaming, owing to the sooty deposit in the form of a bird nest at the end of each tube, the draught is so seriously impeded as to render the boiler practically useless.

The wet-bottom boiler, however, had certain advantages over the dry-bottom, and, when carefully designed, may be fairly satisfactory. Fig. 214 shows the boiler as fitted in the "Rattlesnake," whence it will be seen that the flat tubes connecting the top to the bottom of the fire-box provide a splendid means for circulation, and form a good support to the roof. The fire-bars must be lowered, at the back ends at least, so as to be in the same relative position as in the dry-bottom boiler. The fire-box itself may be divided into two independent parts by a longitudinal water-space, instead of the row of vertical tubes connecting the top and bottom, as in fig. 214. By either method an increase of heating surface is made in the fire-box, and a better means of circulation provided, as also a good supply of water to the top of the fire-box ensured.

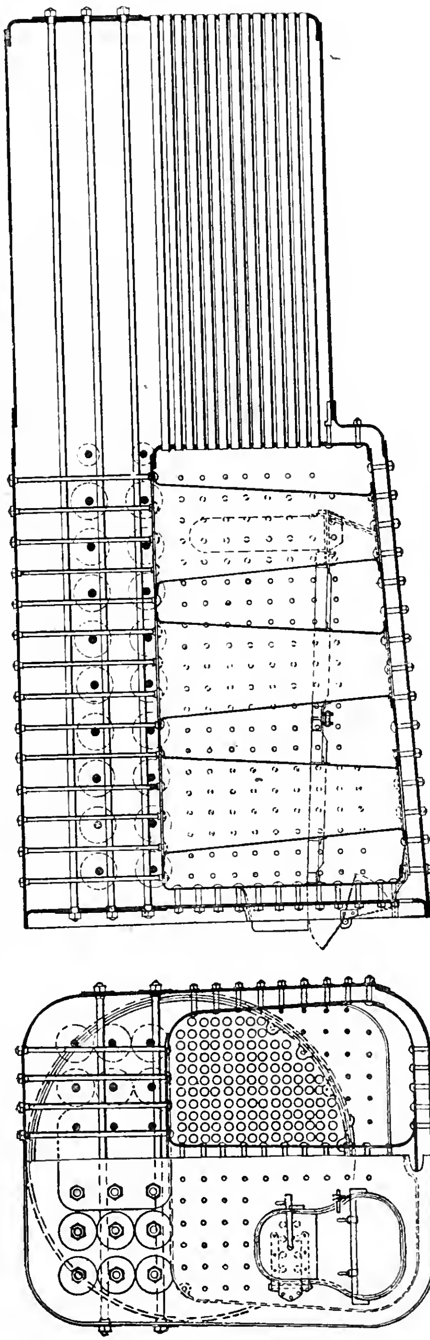


Fig. 214.—Locomotive Boiler. Marine Type (Laird's).

Double-ended Locomotive Boilers.—This form of boiler consists of a fire-box with a tube-barrel at each end, which is to the ordinary locomotive boiler what the double-ended cylindrical boiler is to the single-ended. The firing-holes are on either side, and in the case of a big boiler they will be two in number, so that through the four the grate can be well worked. Moreover.

from their position and size, the opening of them does not so seriously affect the tube-plates as with the ordinary locomotive boiler. This form of boiler is much lighter than the ordinary one, and is convenient in many ways, especially for vessels of limited beam.

Seaton's Locomotive Boiler.—By making the tube barrel of the locomotive boiler a truncated cone instead of a cylinder, and arranging the tubes so that the top rows are nearly horizontal, while the others slant more and more as they approach the bottom, a better means for circulation is provided by the wider spacing of the tubes at the fire-box and the declination of the boiler barrel at the bottom, while, by the inclination of the tubes upwards, a better draught is ensured.

Dimensions of a Boiler.—The amount of grate area is the consideration which chiefly affects the choice of dimensions of boiler, and to a very large extent the number and form of the boilers also are governed by it.

The cylindrical boiler is not elastic in the hands of the designer; to increase the number of furnaces in it its diameter must be increased, which means that both breadth and height are affected. If two furnaces of 40 inches diameter be the limit for a boiler of 10 feet in diameter, that there may be adequate heating surface, and 14 feet is the suitable diameter for three furnaces of 40 inches diameter, the grate bars being of the same length in both cases, the increase in boiler capacity is 96 per cent. for an increase of 50 per cent. of grate. The *smallest* diameter of shell into which three 40-inch furnaces can be fitted so as to give adequate heating surface, is 13 feet 6 inches, which is an increase in capacity of 82 per cent. over the boiler 10 feet in diameter. Four 40-inch furnaces require a shell of at least 16 feet diameter, which means an increase of 156 per cent. capacity to obtain 100 per cent. increase of grate. To arrange four 40-inch furnaces so as to be convenient for stoking, a shell of 17 feet diameter is required, which means an increase of 189 per cent. over the shell of 10 feet diameter; if, instead of increasing the number of *furnaces* by increasing the diameter of shell, the number of *shells* be increased, the space occupied is considerably in excess of the direct ratio of grate areas.

It is true that to some extent increase of grate area may be obtained by increasing the length of furnace, but the *efficiency* of a grate in practice is nearly inversely as its length; for a long grate cannot be nearly so well attended to as a short one, nor is the air supply either under or over the bars so good with a long furnace, since the area of section at the mouth, with the same diameter of furnace, is the same whether the bars be short or long. It is no doubt for this reason that Macfarlane Gray's rule, *that the consumption of coal is very nearly proportional to the diameter of furnace,* is found to be so nearly correct in every-day practice.

Area of Fire Grate.—The area of fire grate required for the evaporation of a certain weight of steam depends on the quantity and quality of the fuel burned on it; the quantity of coal is generally dependent to a large extent on the quality, as may be seen by reference to Table li., and by the draught. It may be assumed that 1 pound of good steam coal will evaporate 10 pounds of water in the ordinary marine boiler, 7 pounds in a locomotive boiler, as fitted to torpedo boats, when not being forced, and 6 pounds when forced to the utmost; also that in the mercantile marine, where the coal is only of moderate quality, 8 to 9 pounds is a fair result, and

TABLE LII.—COMPARISON OF THE DIFFERENT TYPES OF MARINE BOILERS, CONSTRUCTED IN ACCORDANCE WITH THE BOARD OF TRADE OLD RULES FOR A WORKING PRESSURE OF 80 LBS. AND 160 LBS. PER SQUARE INCH.

General Description	Number of Boilers.	Shell.		Furnaces.			Tubes.				Total Heating Surface.	Total Capacity.	Weight. Working Press. 80 lbs.			Weight. Working Press. 160 lbs.		
		Diam.	Length.	No.	Diam.	Length.	No.	Diam.	Length.	Surface.			Boiler.	Water.	Total.	Boiler.	Water.	Total.
		Ft. Ins.	Ft. Ins.		Ins.	Ft. Ins.		Ins.	Ft. Ins.	Sq. feet.	Sq. feet.	Cub. feet.	Tons. Steel.	Tons.	Tons.	Tons. Steel.	Tons.	Tons.
Double-ended boiler, having two combustion chambers, each common to *opposite* furnaces, -	1	12 0	14 0	4	39	5 6	376	$3\frac{1}{4}$	5 6	1725	2000	1582	23·75	20	43·75	35·3	20	55·3
Double-ended boiler, having two combustion chambers, each common to furnaces at *same* end, - - -	1	12 0	14 0	4	39	5 3	376	$3\frac{1}{4}$	5 3	1643	2000	1582	24	20	44	36	20	56
Double-ended boiler, having four combustion chambers, one to each furnace, - - -	1	12 0	14 0	4	39	5 3	376	$3\frac{1}{4}$	5 3	1643	2000	1582	24·2	20·6	44·8	36·4	20·5	56·9
Single-ended boiler, having three combustion chambers, one to each furnace, - - -	1	14 6	9 7	3	42	6 9	274	$3\frac{1}{2}$	6 9	1665	2000	1582	26·15	22	48·15	37·25	22	59·25
Two single-ended boilers, each having two combustion chambers, one to each furnace, -	1	11 0	8 4	4	38	5 10	316	$3\frac{1}{2}$	5 10	1650	2000	1582	30·6	21·4	52	39·2	21·4	60·6

6 to 8 pounds only can be obtained with the coal supplied at some foreign ports. The quantity of coal burnt on a square foot of grate per hour with natural draught is about 20 pounds, under favourable circumstances; with good stoking on short special bars and very good draught as much as 25 pounds may be consumed; but under ordinary circumstances and natural draught, only 15 pounds should be supplied to obtain complete combustion and economical results. With Howden's system of forced draught and warm air 50 pounds per square foot can be burnt, if the coal is fairly good and clean; with closed stokeholes, and an air pressure of $\frac{1}{2}$ inch of water, 30 pounds of good Welsh and 35 of North country coal can be burnt.

From this it will be seen, (1) that the greatest weight of steam evaporated per square foot of grate per hour, under the most favourable circumstances and natural draught, is 10×25, or 250 pounds; (2) that with bad fuel and economical stoking it may be only 6×15, or 90 pounds; (3) that with fairly good fuel and favourable circumstances it may be 9×20, or 180 pounds, and (4) that with fairly good coal and careful stoking about 150 pounds may be expected. In practice, therefore, for short trial trips with picked Welsh coal and picked stokers, calculations may be based on an evaporation of 250 pounds; for mail steamships using good English coal, calculations should be based on an evaporation of 150 pounds for natural draught, and for forced draught from 270 to 400, depending on the air pressure; and if a ship is going to trade in the East or localities where inferior coal only can be obtained, the boilers should be designed on the assumption of an evaporation of only 100 pounds of water per square foot of grate natural, and 200 to 280 forced draught.

If the weight of steam required per hour for a given engine be calculated and divided by one of these numbers, the result will be the number of square feet required.

If the draught be increased by artificial means to a still higher intensity, as is the case with torpedo boats, etc., the quantity of fuel consumed per square foot of grate may be as high as 100 pounds per hour, with an air pressure of 6 inches in the stokehole; and 50 pounds with only 2 inches, the corresponding evaporations being 570 pounds and 350 pounds per square foot of grate.

Consumption of Fuel.—The consumption of fuel per I.H.P. per hour for engines working at full power was 4 pounds, with surface-condensing expansive engines, using steam of 30 pounds pressure above the atmosphere; $3\frac{1}{4}$ to $3\frac{1}{2}$ pounds with similar engines of best make and large size; $2\frac{3}{4}$ pounds with compound engines when forced, and $2\frac{1}{4}$ to $2\frac{1}{2}$ pounds when of moderate size and working at two-third power; $2\frac{1}{4}$ pounds with compound engines of moderate size and as generally fitted in the mercantile marine when working at full speed; 2 pounds with the best compound engines well designed and carefully worked at sea full speed; $1\frac{3}{4}$ pounds with large compound engines when working at sea full speed with best fresh Welsh coal: $1\frac{1}{2}$ pounds with good triple-expansion engines using English and Welsh coal of really good quality, and $1\frac{3}{4}$ pounds when ordinary good steam coal is used; $1\frac{1}{4}$ to $1\frac{1}{2}$ pounds with triple- and quadruple-expansion engines using steam at 200 to 180 pounds pressure; the consumption of water with these latter engines being about 14 to 15 pounds; the consumption in torpedo boats with compound engines was $3\frac{1}{2}$ to 4 pounds when working nearly full speed, and $2\frac{1}{2}$ pounds with triples in Destroyers. In H.M. Navy using Welsh coal and an air pressure of $\frac{1}{2}$ inch of water the consumption is 1·75 pounds including auxiliaries; with Howden's system in the mercantile marine 1·3 to 1·4 pounds of good Welsh coal with large engines and 1·35 to 1·5 with smaller is general. With superheaters for the steam in addition to the hot air of the Howden and Ellis-Eaves systems the consumption is still lower, and generally does not exceed 1·3 pounds per I.H.P. per hour for main engines only.*

Assuming the consumption of coal to be $1\frac{1}{2}$ pounds per I.H.P. per hour, and the grate to burn 15 pounds per square foot, there should be 0·1 square foot of grate per I.H.P. If the sea full speed I.H.P. of a merchant ship be multiplied by 0·1 it will give sufficient grate area for that power.

That is, the grate area required for that power = I.H.P. ÷ 10.

On trial trips with good coal, natural draught, and the engines working at full speed, the triple-compound engine will develop 14 I.H.P. per square foot of grate; hence, one-fourteenth of a square foot per I.H.P. may be taken as the proper allowance in designing furnaces for engines to develop a certain power on favourable occasion, and one-thirtieth with Howden fittings

As the sea full speed power is usually, on long voyages, about three-fourths that developed on a trial trip, the proportion of $\frac{4}{3} \times \frac{1}{14}$, or 0·095 of a square foot per I.H.P. developed at sea, corresponds with that given above. But with the higher funnels of the larger steamships even better results are obtained, and it is found now that even 0·08 square foot per I.H.P. as developed at sea is sufficient for the main engines only; for the auxiliaries and domestic purposes a special addition must be made in proportion to their needs.

Heating Surface.—Strictly speaking, all surfaces exposed to heat which are capable of absorbing, and their bodies of transmitting, that heat to the water or steam are heating surfaces; but technically only certain parts are reckoned as *effective* heating surface, and the aggregate of such surfaces is called the *total heating surface.* The surface of the upper half of the furnace, or the part above the level of the fire-bars, that of the combustion chamber above the level of the bridges and back slabs, including the actual surface of the back-tube plates, are reckoned as the effective heating surface of furnaces and chambers, and are stated separately, chiefly on account of the metal forming them being three to four times the thickness of the tubes. The surface of the tubes measured externally—that is, the area obtained by multiplying the external circumference by the length *between* the tube plates, is called the *tube surface.*

The Admiralty reckon tube surface by taking the area obtained by multiplying the external circumference by the length *over* the tube plates; and in reckoning the total heating surface, the surface of the back tube plates is omitted. The calculation is in this way simplified, and the *total heating surface* is practically the same as if calculated strictly, for the surface of the parts of the tubes covered by the plates is very nearly the same as that of the back tube plates.

The *front* tube plates should be, and usually are, omitted in calculating the total heating surface, as they cannot be considered as effective.

The *amount* of total heating surface must depend on the quantity and quality of the fuel burnt on the grates in a fixed time—that is, on the quantity of heat generated in a unit of time, and also on the quality of the

* Experiments on ships of large size fitted with Robinson superheaters showed the consumption of good coal with triples to be 1·4, and with Howden's fittings 1·18. Quadruples 1·15 lbs. per hour.

surface, etc. But since a grate may at some time have to burn the best of fuel, and the engine to have the advantage, the total heating surface should be adequate for such an occasion. If, however, no increase in engine power is required, the damper must be partly closed or a portion of the grate bricked up; the latter is by far the better plan on economic grounds, as the brighter the fire the more effective is the combustion, and the consumption correspondingly decreased.

Tube Surface.—When possible, there should be 1·33 square feet of iron or steel tube for each pound of coal burnt per hour—that is, in the ordinary marine boiler there should be about 27 square feet of iron tubes per square foot of grate; with brass and copper tubes it was usual to provide about 25 per cent. less surface.

Since on trial trip with compound engines 10 I.H.P. were usually developed per square foot of grate, there should be 2 square feet of brass, and 2·7 square feet of iron tubes per I.H.P. developed on trial trip, or 2·66 and 3·6 square feet respectively per I.H.P. developed at sea. And since with triple-expansion engines and natural draught as much as 14 I.H.P. are developed from a square foot of grate, there need only be 1·93 square feet of iron tube surface per I.H.P. in the boilers for them. When weight of boiler is of as much consideration as economy of fuel 1·75 square feet of tube surface per I.H.P. is sufficient with triple-expansion engines. With forced draught by closed stokeholes and 30 to 40 pounds of coal are consumed the heating surface per foot of grate must be accordingly increased; the tube surface should be, therefore, still 1·33 feet per pound of fuel, and, consequently, 40 to 53 times the grate area.

Total Heating Surface must be governed as to amount by the ability to produce steam, and the weight of steam per horse-power required by the engine, etc. In general practice with modern marine boilers of good design the following holds good for the production of steam per square foot per hour under favourable conditions as to the state of inner and outer surfaces of tubes, the draught, air temperature and humidity, etc., viz.:—

(*a*)	Best class of naval water-tube boiler,	highly forced draught,	13 lbs.
(*b*)	,, ,, ,,	moderately ,,	10 ,,
(*c*)	Ordinary water-tube boiler,	lightly ,,	7·5 ,,
(*d*)	Cylindrical boilers,	moderately ,,	9·0 ,,
(*e*)	,,	ordinary draught,	6·5 ,,

The steam consumption on various ships, including that required for auxiliaries of every kind as well as the demands for domestic purposes, may be taken as follows:—

Description of Ship and Service.	Main Engine.	Auxiliary, &c.	Margin Supply.	Total Consumpt.
Cargo steamer, general service, quadruple compound engine,	14·0	0·70	1·40	16·1
,, ,, triple compound engine,	14·7	0·74	1·47	16·9
,, ,, geared turbines,	12·5	0·85	1·25	14·6
Passenger express steamers, Atlantic, turbines,	12·5	2·50	1·25	16·25
,, ,, Foreign, reciprocators,	13·5	1·50	1·35	16·35
,, ,, Foreign, turbines, geared,	12·5	1·50	1·25	15·25
,, ,, Cross-Channel, turbines,	13·0	1·65	1·30	15·95
Naval ships, Battleships and Cruisers, turbines,	13·5	2·00	1·35	16·85
,, Destroyers, Scouts, etc., turbines,	14·0	2·00	1·00	17·00

If these figures be taken in conjunction with the amounts evaporated as above as a basis from which it may be deduced that the ordinary cylindrical

TABLE LIII.—TOTAL HEATING SURFACE IN PRACTICE (POWER IS MAXIMUM DEVELOPED ON 2-HOUR TRIAL).

Kind of Boiler.	Ship.		Kind of Engines.	Rate of Total Heating Surface.	Naval.		Mercantile.	
	Description of.	Service on.			Natural Draught.	Forced Draught.	Natural Draught.	Forced Draught.
					Square Feet.	Square Feet.	Square Feet	Square Feet.
Cylindrical.	Ordinary.	General.	Any.	Per lb. of coal per hr.	1·33	1·10	1·60	1·33
,,	,,	,,	,,	Per lb. of oil per hr.	1·82	..	2·20	..
,,	,,	,,	,,	Per sq. foot of grate.	27·0	30 to 35	30 to 33	40 to 60
Water-tube.	Naval & Express	,,	,,	,, ,,	30 to 36	40 to 60	..	..
,,	Battleship and Cruisers.	,,	Turbines.	Per S.H.P.	2·10	1·87	..	..
,,	Scouts & T.B.Ds.	,,	,,	,,	..	1·42	..	..
,,	,, ,,	,,	Triples.	Per I.H.P.	..	2·25	..	..
Cylindrical.	Express passenger.	Atlantic.	Quadruple.	,,	..	..	2·56	2·24
,,	,,	,,	Turbine.	Per S.H.P.	..	..	2·17	1·97
,,	Passenger and cargo.	General.	,,	,,	..	..	1·88	1·75
,,	,, ,,	,,	Quadruple.	Per I.H.P.	..	..	2·11	1·91
,,	,, ,,	,,	Mixed.	Per S.H.P.	..	..	1·90	1·70
,,	Express passenger.	Cross-Channel.	Turbine.	,,	..	..	1·60	1·50
,,	,, ,,	,,	Triples.	Per I.H.P.	..	..	1·91	1·80
,,	Cargo (tramp).	General.	,,	,,	..	..	2·82	2·50
,,	,, ,,	,,	Quadruple.	,,	..	..	2·68	2·40
,,	,, ,,	,,	Grd. trbs.	Per S.H.P.	..	..	2·45	2·20

boiler on service should supply 8 lbs. on short service with forced draught, and 7 lbs. on long service with good draught as with Howden's apparatus, or without but with high chimneys, and 6 lbs. for ordinary cargo ships. The express steamer with water-tube boilers and forced draught should have 10 lbs. on short service and 9 lbs. on long. The Scout and Destroyer will have 12 lbs., and the Cruiser and Battleship 9 lbs. The merchant ship on long service usually proceeds with the engines at three-quarter power; hence the evaporation then will be 75 per cent. of these allowances and the efficiency should be 0·80. The express cross-channel steamer will usually work with 85 per cent. of the power, so that the demand per square foot will then be 8·5 lbs. Naval ships proceed at very varying speeds.

The amount of total heating surface which a cylindrical boiler may have is given approximately by the following rule:—

(1) Total heating surface = (diameter of shell in feet)^2 $\times$ (length of tubes in feet + 3).

Example.—What amount of heating surface should be found in a boiler 12 feet diameter, the tubes being 7 feet long?

Total heating surface = $12^2 \times (7 + 3)$, or 1,440 square feet.

Example.—What heating surface should a double-ended boiler contain whose diameter is 10 feet, and the length of tubes 5·5 feet, the combustion chambers being common to opposite furnaces?

Here total heating surface = $10^2 \times 2 \times (5{\cdot}5 + 3)$, or 1,700 square feet.

If the combustion chambers are the same as in a single-ended boiler, the quantity will be the same as in *two* single-ended boilers of the same diameter and length of tubes.

Efficiency of Heating Surface.—The rate of transmission of heat from the fuel and hot gases in a boiler to the water on the other side of the metal separating them depends on the differences of temperature, the thickness of the plate and its material, but chiefly on the state of the receiving and transmitting surfaces. Some few years ago elaborate and careful experiments were made by Sir John Durston and the late Mr. Blechynden independently, which demonstrated that, when the plate is clean on both sides, there is not a great difference between the temperature on the sides of the plate itself, the dry side being, in practice, only 67° to 86° F. hotter. If, however, the transmitting side is dirty with scale, etc., it may be very much greater; and, with greasy mud, it was over 500°. That a slight layer of soot or tarry matter prevented the dry side from getting very hot was demonstrated, as also that, *until the plate* is raised to a certain temperature, the actual or burning gas flame does not come into real contact. Further, that the *thickness* of plate when clean is not of much consequence, and is less so when dirty; it is, however, of some importance as being a regulator of the transmission of heat. Neglecting thickness altogether, it was found that the efficiency varies as the square of the differences of temperature, and the following formula may be taken as correct:—

$$Q = \frac{(T - t)^2}{a},$$

Q being the units of heat per hour per square foot of heating surface.

T the temperature of the hot gas.

t ,, ,, water.

a is a factor which Rankine valued at 160 to 200; Blechynden at 40 for tubes *when cleaned*, and 50 for plates $\frac{5}{8}$ inch thick *when clean*.

There is reason to think that Blechynden is nearer the truth than Rankine for modern boilers worked on modern conditions.

There is every reason to suppose that the efficiency is seriously affected by such inequalities of surface as permit of disturbing the flow of hot gases over it and thereby bringing really more heat in contact.

Efficiency of Boilers.—Generally, it may be taken that the efficiency of a boiler is best gauged by taking the water evaporated by a pound of coal from and at 212° F. and divide it by the amount such fuel could evaporate if all its heat were used for the purpose. In practice it is, of course, impossible to utilise the whole, or even nearly the whole, heat of combustion. It is, moreover, not to be expected of certain boilers that the efficiency so measured can be very high, but it is necessary to compare them with others of their kind as well as with boilers generally.

A boiler for express service, such as that of a torpedo boat or a short-distance passenger ship, must be light, and the measure of its efficiency for this purpose will be gauged better by knowing how many pounds of steam it can produce per square foot of heating surface per hour. Now, weight is always a factor of some value on shipboard. If, therefore, the amount be multiplied by the number of pounds of water it evaporates (from and at 212° in each case), or the number of pounds of steam it produces per pound of standard fuel, the result must express the general value of the boiler for steamship purposes. The weight of steam supplied per hour for each ton weight of boiler and appurtenances, including water, is really a practical criterion of the efficiency of a boiler for marine purposes.

Area through Tubes.—The sectional area through the tubes, or that area through which the hot gases and smoke pass from the combustion chamber to the funnel, should not be less than one-seventh the area of grate with the natural draught, and is usually about one-fifth. Too large an area produces a low velocity, which permits a deposit of soot and ash to form, with the consequent reduction of evaporative efficiency. Too small an area checks the draught, especially when the surfaces have become dirty. With forced draught the area through tubes can be smaller if necessary; and when it is more than one-seventh the grate (for natural draught) it is advantageous to have retarders in the tubes. These spiral strips not only prevent a too rapid flow of gas but force the hot current over every portion of the tube's circumference.

Capacity of Boiler Shell.—To contain the requisite heating surface, and to leave sufficient steam space, the boiler shell should contain 3 cubic feet per I.H.P. for the mercantile marine, and 2·5 cubic feet for the Navy and other service which has equally quick-running engines, when made for compound screw engines, and 2 and 1·6 respectively for triple engines; when for paddle engines it should be larger. In the mercantile marine a slightly larger allowance is sometimes made, especially when for steamers making long voyages. Since the *volume* of steam produced at a pressure of 210 lbs. is only about a third that at 70 lbs. when the same weight is used, the steam space for boilers working at the high pressures now obtaining may be considerably less than was usual. Good results can be got now with an allowance of $1\frac{3}{4}$ to 2 cubic feet of boiler per I.H.P. for natural draught, and $1\frac{1}{4}$ to $1\frac{1}{2}$ for forced draught. In fact, the steam space must bear a relation to the high-pressure cylinder capacity in every case; the intermittent demand of the single high-pressure cylinder of a paddle engine affects the equilibrium of the boiler very differently from that of the two high-pressure cylinders of a twin screw ship running at four times the revolutions, but using the same quantity of steam.

* **Steam Space.**—The top row of tubes in a cylindrical boiler should be not less than 0·28 of the diameter of the shell from the top. The tubes are sometimes placed higher, but there is then risk of priming owing to contraction of water-surface area as well as to the conformation of the sides. Priming is often due rather to contracted area of water surface than to small steam space, although the latter is generally set down as the cause when the tubes are high. If the water surface is so contracted that little or no part of the boiler where there may be down currents lies immediately under it, priming is sure to ensue.

The capacity of the steam space depends on the quantity of steam used in a fixed time, and on the number of periods of supply to the engines in that time. The effect of admission to the cylinders is to reduce the pressure in the steam space; if that reduction in pressure be sensible there will be an augmentation of ebullition at that period; if the reduction in pressure is serious there will be excessive ebullition, resulting in *priming*. For this reason it is that a slow-moving paddle engine, which takes its steam in a series of gulps, requires to have boilers with much larger steam space than is sufficient for fast-running screw engines using the same weight of steam.

There should be 0·8 of a cubic foot of steam space per I.H.P. for a slow-running paddle engine, and 0·65 of a cubic foot per I.H.P. for compound screw engines, and as low as 0·55 of a cubic foot for fast-running mercantile and naval screw compound engines. Boilers for triple- and quadruple-expansion engines may have 25 per cent. less steam space than this with natural draught, and 50 per cent. less with quick-running ones and forced draught; while with turbines the space is even less. The amount of steam space in a boiler is not dependent on the *weight* of steam used per stroke. but on the *volume*, and for that reason a boiler constructed for, say, 200 lbs. pressure does not require so much steam space as a similar boiler constructed for only 75 lbs. pressure; for if both boilers have the same grate area and heating surface, they will evaporate the same *weight* of steam, but the volumes will be as 3 to 8 nearly. If these two boilers supply steam at the same rate to engines running at the same number of revolutions, their steam spaces may be as 3 to 8 nearly.

It is no doubt on account of the high pressure and large number of revolutions of engines, together with the shaking when running, that a locomotive boiler works so well without priming, in spite of the small steam space and contracted water surface.

Area of Uptake and Funnel Sections.—Although in practice the funnel is often designed to suit the general appearance of the ship, and it is also found that, whereas some engineers prefer a small high funnel, there are others who, for some other reasons, resort to the practice of making the funnel short and of large diameter; still there is undoubtedly a certain diameter and a certain height that will give the best result, and that cannot be determined from external considerations.†

* When turbines take the steam the space does not require to be so large as obtains with reciprocators.
† Consult *Index* for other information.

In the Navy with natural draught the funnel is usually made with a sectional area equal to *one-eighth the area of the grate* (natural draught size). In the mercantile marine a somewhat larger funnel usually obtains, the area being from one-fourth to one-sixth that of the grate; in general practice a funnel whose sectional area is one-fifth to one-sixth that of the grate, and whose top is at least 40 feet from the level of the grate, will give a very good result. The objections to a large funnel, beyond that of space occupied and cost, are resistance to the wind and large surface exposed to the cooling action of both wind and water, whereby the hot column within is partially cooled, and the draught thereby checked. On the other hand, a small funnel is liable to become excessively hot, and when the fires are freshly charged to become choked with smoke, and at all times it tends to check the draught. The funnel of a warship may be small, because it is so seldom that the boilers are urged to the utmost, and it must be as small as possible for obvious reasons. When the draught is forced either by a blast or by other artificial means, the funnel may be short, and of comparatively small diameter. The area at the base of a locomotive is seldom more than one-tenth the area of fire-grate, and often as small as one-twelfth. The size, for appearance sake, is now got by making the outer casing of the required form.

Area of Funnel Section should be such that (1) for natural draught Merchant service condition there are 1·25 square inches for each pound of fuel consumed per hour, or, say, 1·88 square inches per I.H.P. of trial trip.

(2) For Naval service and short Express service conditions there should be 1·7 square inches with assisted draught, and 1·25 with forced draught per I.H.P. of trial trip, or 0·9 square inch per pound of coal burned per hour.

Diameter funnel in inches = $\sqrt{\text{I.H.P.} \times \text{F.}}$
For ordinary Merchant steamer F = 2·38 natural draught.
" Naval or Express " F = 2·16 assisted "
" " " F = 1·66 forced "

TABLE LIII*a*.—CAPACITY OF FUNNELS FOR COAL BURNT IN LBS. PER HOUR.

Diameter of Funnel.	Height of Funnel above Dead Plates in Feet.													
	20	25	30	35	40	50	60	70	80	90	100	110	120	140
2·0	167	187	207	224	239	264	288	311	333	..	..	..	..	..
2·5	261	292	320	346	370	413	453	489	522	..	..	..	..	..
3·0	380	421	464	500	535	595	653	705	753	..	..	..	..	..
3·5	515	575	630	681	728	813	891	963	1,028	..	..	..	..	..
4·0	..	750	822	888	948	1,061	1,161	1,254	1,341	1,422	..	..	..	..
5·0	..	..	1,282	1,385	1,480	1,654	1 812	1,956	2,088	2,221	..	..	..	..
6·0	..	..	..	..	2,140	2,380	2,609	2,820	3,012	3,196	3 366	..	..	..
7·0	..	..	..	..	2,916	3,252	3.564	3,852	4,112	4,364	4,580	..	..	..
8·0	..	..	..	..	..	4,228	4,644	5,016	5,448	5,688	5,980	6,283	..	..
9·0	..	..	..	..	..	..	5,884	6,356	6,788	7,204	7.592	7,961	8,310	8,980
10·0	..	..	..	..	..	..	7,248	7,824	8,352	8,885	9,360	9,820	10.250	11,070
11·0	..	..	..	..	..	..	..	9,466	10,110	10,730	11,310	11,840	12.380	13,380
12·0	..	..	..	..	..	..	..	11,280	12,050	12,700	13,460	14.120	14.750	15,930

TABLE LIV.—PARTICULARS OF SOME MARINE BOILERS MADE OF STEEL (UNDER OLD RULES).

Rules on which Constructed.	Shell			Working Pressure.	Furnaces.			Number of Combustion Chambers.	Tubes.				Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
	Diameter.	Length.	Thickness.		Number.	Diameter.	Thickness.		Number.	Diameter.	Length.	Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
	Ft. In.	Ft. In.	In.	Lbs.		In.	In.			In.	Ft. In.	Sq.Ft.	Sq Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
Board of Trade, - - - -	14 9	16 6	$1\frac{11}{32}$	160	6	42	$\frac{9}{16}$	1	336	$3\frac{1}{2}$	6 6	3260	3750	58·2	32·0	90·2	1·55	0·85	2·4	64·4
Lloyd's and Board of Trade,	14 3	16 0	$1\frac{5}{16}$	160	6	$38\frac{1}{2}$	$\frac{19}{32}$	3	590	$3\frac{1}{4}$	6 3	3074	3620	57·75	30·6	88·35	1·60	0·84	2·44	62·7
,, ,, ,	14 0	17 0	$1\frac{1}{4}$	160	6	40	$\frac{35}{64}$	3	466	$3\frac{1}{2}$	6 9	2840	3375	57·10	30·0	87·10	1·69	0·88	2·57	59·1
,, ,, ,,	13 3	16 3	$1\frac{5}{32}$	150	6	$34\frac{1}{2}$	$\frac{31}{64}$	3	468	$3\frac{1}{4}$	6 $4\frac{1}{2}$	2490	3000	45·0	26·0	71·0	1·50	0·86	2·36	66·7
,, ,, ,,	13 6	16 5	$1\frac{7}{32}$	165	6	40	$\frac{9}{16}$	3	436	$3\frac{1}{4}$	6 4	2350	3000	51·0	26·0	77·0	1·70	0·86	2·56	58·9
,, ,, ,,	13 0	16 0	$1\frac{5}{32}$	154	4	43	$\frac{18}{32}$	2	464	$3\frac{1}{4}$	6 3	2417	2810	42	24·75	66·75	1·49	0·88	2·37	66·2
,, ,, ,.	12 0	16 3	$1\frac{1}{8}$	150	4	39	$\frac{9}{16}$	2	364	$3\frac{1}{2}$	5 $5\frac{1}{2}$	1775	2260	38	28·2	66·2	1·68	1·24	2·92	59·5
Board of Trade, - - -	11 9	15 0	1	150	4	40	$\frac{1}{2}$	2	296	$3\frac{1}{4}$	6 0	1740	2100	31·4	20·3	51·7	1·48	0·97	2·46	66·8
Lloyd's, - - - - -	11 2	18 3	$1\frac{1}{32}$	150	4	38	$\frac{1}{2}$	1	208	$3\frac{1}{2}$	7 0	1310	1685	34·3	21·2	55·5	2·03	1·30	3·33	49·1
,, and Board of Trade,	12 3	16 6	$1\frac{1}{8}$	160	4	38	$\frac{1}{2}$	1	344	$3\frac{1}{2}$	6 7	2033	2380	39·7	22·55	62·25	1·67	0·94	2·61	60·0
,, ,, ,,	14 6	10 6	$1\frac{5}{16}$	160	3	42	$\frac{5}{8}$	3	232	$3\frac{1}{2}$	8 0	1820	2250	40·0	23·0	63·0	1·77	1·02	2·80	56·3
,, ,, ,,	14 3	9 3	$1\frac{5}{16}$	160	3	$38\frac{1}{2}$	$\frac{19}{32}$	3	295	$3\frac{1}{4}$	6 3	1537	1910	35·0	17·5	52·5	1·83	0·91	2·74	54·6
,, ,, ,,	14 0	10 6	$1\frac{9}{32}$	155	3	42	$\frac{9}{16}$	3	226	$3\frac{1}{2}$	7 $3\frac{1}{2}$	1484	1870	37·2	19·5	56·7	1·99	1·04	3·03	50·0
Board of Trade, - - -	13 9	10 6	$1\frac{3}{32}$	142	3	38	$\frac{7}{16}$	3	192	$3\frac{3}{4}$	7 3	1340	1730	31·2	20·2	51·4	1·8	1·16	2·96	55·4
,, - - - -	13 0	11 0	$1\frac{9}{32}$	150	2	48	$\frac{19}{32}$	2	176	$3\frac{3}{4}$	7 6	1270	1590	31·3	17·2	48·5	1·96	1·09	3·05	50·1
,, and Lloyd's,	13 0	10 6	$1\frac{1}{8}$	150	3	36	$\frac{1}{2}$	3	152	$3\frac{3}{4}$	7 0	1024	1360	30·4	16·5	46·9	2·24	1·21	3·45	44·7
,, - - - -	12 3	9 6	$1\frac{3}{8}$	200	2	42	$\frac{21}{32}$	1	192	$3\frac{3}{4}$	6 6	1045	1300	27·0	13·0	40·0	2·07	1·00	3·07	48·1
,, and Lloyd's,	12 0	9 $1\frac{1}{2}$	$1\frac{1}{8}$	150	2	39	$\frac{9}{16}$	1	182	$3\frac{1}{2}$	6 $1\frac{1}{2}$	1000	1240	23·1	13·04	36·14	1·86	1·05	2·91	53·7
,, - - - -	11 9	11 0	1	150	2	46	$\frac{17}{32}$	1	128	$3\frac{3}{4}$	8 0	990	1250	22·6	14·0	36·6	1·80	1·12	2·92	55·3
,, and Lloyd's,	11 9	9 6	$1\frac{3}{32}$	150	2	39	$\frac{9}{16}$	2	142	$3\frac{1}{2}$	6 6	830	1085	23·25	12·2	35·45	2·14	1·12	3·26	46·7
Lloyd's, - - - -	11 6	9 6	$\frac{15}{16}$	140	2	40	$\frac{1}{2}$	2	168	$3\frac{1}{4}$	6 6	913	1145	20·0	10·7	30·7	1·74	0·93	2·68	57·2
Board of Trade, - - -	10 6	9 3	$\frac{7}{8}$	150	2	38	$\frac{1}{2}$	2	118	$3\frac{1}{4}$	6 3	614	810	16·2	10·0	26·2	2·0	1·23	3·23	50·0
Lloyd's, - - - - -	10 6	9 3	$\frac{7}{8}$	150	2	37	$\frac{5}{8}$	1	110	$3\frac{1}{2}$	6 0	642	800	16·2	9·0	25·2	2·02	1·12	3·15	49·4
,, - - - - -	9 0	8 6	$\frac{27}{32}$	150	2	32	$\frac{7}{16}$	2	100	$3\frac{1}{4}$	6 0	500	655	12·6	6·5	18·6	1·92	1·00	2.92	52·0
,, - - - - -	9 3	9 $1\frac{1}{2}$	$\frac{13}{16}$	155	2	31	$\frac{19}{32}$	1	104	$3\frac{1}{4}$	6 3	541	692	13·2	8·0	21·2	1·90	1·15	3·05	52·6
Board of Trade, - - -	8 6	8 6	$\frac{7}{8}$	150	2	30	$\frac{1}{2}$	2	104	3	5 11	473	600	11·5	6·68	18·18	1·93	1·11	3·04	52·2

TABLE LV.—LEADING PARTICULARS OF SOME BOILERS MADE UNDER THE NEW RULES.

Designation.	Made in Accordance with the Rules of	Working Pressure.	Shell.			Furnaces.			Number of Combustion Chambers.	Tubes.				Total Heating Surface.	Weight of			Weight per 100 feet of Total Heating Surface.			Sq. Ft. Total H.S. per Ton of Boiler.
			Diameter.	Length.	Thickness.	Number.	Diameter.	Thickness.		Number.	Diameter.	Length.	Surface.		Boiler.	Water.	Total.	Boiler.	Water.	Total.	
		Lbs.	Ft. In.	Ft. In.	In.		In.	In.			In.	Ft. In.	Sq.Ft.	Sq.Ft.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	
H D	Board of Trade, -	190	16 5	17 10	$1\frac{17}{32}$	8	$40\frac{3}{4}$	$\frac{19}{32}$	...	848	$2\frac{3}{4}$	7 2	4293	5112	83·0	47	130	1·62	0·92	2·54	61·6
W F	,, ,, and Lloyd's, - -	180	16 4	17 6	$1\frac{1}{2}$	8	$40\frac{5}{8}$	$\frac{9}{16}$	...	840	$2\frac{3}{4}$	7 $0\frac{1}{2}$	4170	4572	79·5	45·5	125	1·74	0·99	2·73	57·5
P R	Board of Trade, -	210	15 6	19 6	$1\frac{13}{32}$	6	44	$\frac{11}{16}$	...	852	$2\frac{5}{8}$	7 $10\frac{5}{8}$	4540	5210	83·0	46·75	129·7	1·59	0·91	2·50	62·7
W N	,, ,, - -	192	15 3	18 6	$1\frac{15}{32}$	6	$43\frac{1}{2}$	$\frac{5}{8}$	...	826	$2\frac{5}{8}$	7 6	4180	4839	73·75	42·75	116·5	1·52	0·88	2·40	65·6
B T	,, ,, - -	214	14 5	19 10	$1\frac{9}{16}$	6	$40\frac{11}{16}$	$\frac{21}{32}$	...	740	$2\frac{1}{2}$	7 11	3760	4415	75·5	41·75	117·25	1·71	0·95	2·66	58·5
C A	,, ,, and Lloyd's, - -	200	14 3	19 6	$1\frac{13}{32}$	6	42	$\frac{5}{8}$	3	730	$2\frac{1}{2}$	7 $7\frac{1}{2}$	3578	4575	68·5	40·5	109·0	1·50	0·88	2·38	66·8
I D	Board of Trade, -	216	14 2	19 6	$1\frac{9}{16}$	6	$40\frac{11}{16}$	$\frac{21}{32}$	...	612	$2\frac{3}{4}$	8 0	3460	4071	72·5	40·5	113·0	1·78	1·00	2·78	56·1
C P	Lloyd's, - - -	180	12 9	18 0	$1\frac{3}{16}$	6	38	$\frac{17}{32}$	3	616	$2\frac{3}{4}$	6 6	2826	3415	41·0	30·1	71·1	1·20	0·88	2·08	83·3
P E	Board of Trade and Lloyd's, - -	190	11 6	18 9	$1\frac{5}{32}$	4	42	$\frac{19}{32}$	4	496	$2\frac{3}{4}$	7 $1\frac{5}{8}$	2494	3000	43·2	22·5	65·7	1·44	0·75	2·19	69·4
P G	,, ,, ,,	180	15 0	11 9	$1\frac{11}{32}$	3	45	$\frac{19}{32}$	3	409	$2\frac{3}{4}$	8 0	2355	2860	45·0	26·0	71·0	1·57	0·91	2·48	63·5
O O	,, ,, ,,	200	14 3	11 6	$1\frac{5}{16}$	3	44	$\frac{5}{8}$	3	238	$3\frac{1}{2}$	7 9	1667	2138	40·6	19·0	59·6	1·90	0·89	2·79	52·6
D N	Board of Trade, -	160	14 3	11 6	$1\frac{3}{16}$	3	44	$\frac{17}{32}$	3	240	$3\frac{1}{2}$	8 $1\frac{1}{2}$	1756	2232	37·85	19·55	57·4	1·70	0·88	2·58	56·3
Z O	,, ,, and Lloyd's, - -	200	13 0	11 0	$1\frac{5}{16}$	3	38	$\frac{9}{16}$	3	234	$3\frac{1}{4}$	7 6	1492	1907	34·8	17·2	52·0	1·82	0·90	2·72	54·8
S A	Lloyd's, - - -	180	13 0	10 6	$1\frac{1}{16}$	3	46	$\frac{19}{32}$	3	176	$3\frac{1}{2}$	6 9	1067	1335	26·5	16·5	43·0	1·92	1·24	3·16	50·4
D O	Board of Trade and British Corporation,	200	11 6	11 3	$1\frac{1}{8}$	2	40	$\frac{19}{32}$	2	172	$3\frac{1}{4}$	7 7	1088	1327	27·3	13·8	41·1	1·91	1·04	2·95	50·0 {serve tubes
L Z	Lloyd's, - - -	200	11 0	9 6	1	2	$37\frac{3}{4}$	$\frac{9}{16}$	2	132	$3\frac{1}{4}$	6 3	686	880	18·85	10·5	29·35	2·14	1·19	3·33	46·7
B Y	,, - - -	160	11 0	9 6	$\frac{15}{16}$	2	36	$\frac{1}{2}$	2	146	$3\frac{1}{4}$	6 3	761	1000	17·6	10·2	27·8	1·76	1·02	2·78	56·8
D N	,, - - -	200	10 0	9 6	1	2	35	$\frac{5}{8}$	2	118	$3\frac{1}{4}$	6 $5\frac{1}{2}$	650	820	17·3	8·75	26·05	2·11	1·67	3·78	47·4
C O	,, - - -	180	10 0	9 6	$\frac{15}{16}$	2	35	$\frac{9}{16}$	2	118	$3\frac{1}{4}$	6 $5\frac{1}{2}$	650	820	16·4	8·75	25·15	2·00	1·67	3·67	50·0
L N	Bureau Veritas, -	180	8 9	9 9	$\frac{27}{32}$	2	32	$\frac{11}{16}$	1	82	$3\frac{1}{4}$	6 9	561	600	13·4	7·45	20·85	2·23	1·24	3·47	44·8

CHAPTER XXIII.

WATER-TUBE BOILERS.

Such boilers as are composed wholly, or nearly wholly, of small tubes with the water within them are called "water-tube" boilers. Singular to say, the same causes that brought this type of boiler to the front in our time operated over eighty years ago on the minds of the engineers of that day, to produce many inventions and numerous patents, some of which are the prototypes of the most successful boilers of their kind in use at the present time. Others, while displaying a very considerable knowledge of the laws governing the circulation, so necessary for the success of every boiler, and indicating a mechanical knowledge of the highest order, were almost impossible of manufacture then, and have not survived, owing to their inferiority to other designs. In the third decade of the nineteenth century, when the railway locomotive had been tried and found to be a success for traction, many minds were engaged on designing and perfecting steam-driven road carriages; in other words, they were attempting to substitute for the stage coach and the four-horsed travelling carriage a similar conveyance to go on the highways with a steam engine as the motive power. Now, on an iron track the weight of the engine was not of very serious moment; but to run successfully on a turnpike road, without doing such damage as would prohibit its use, the machinery was necessarily as light as possible, because it was carried on the same four wheels as had hitherto supported only the passengers and luggage. Naturally, therefore, attention was drawn to the boiler as being by far the heaviest part of the machinery, and of that part the water was no inconsiderable item; hence, engineers availed themselves of their knowledge of the properties of the cylinder to devise boilers on the tubulous system. Of them, by far the most ingenious were those patented by the well-known physician-engineer Goldsworthy-Gurney in 1827; the best of which consisted of a horizontal cylindrical drum on the top, and a somewhat smaller drum at the bottom, these two being connected by down-cast pipes at the ends, and by up-cast pipes at the sides, the latter being in the shape of the letter **S,** and each pair together forming the figure 8 (see fig. 215). This boiler is interesting, as being the fore-runner and type of the Thornycroft (fig. 230) and other similar boilers of to-day. There were, besides this, other forms almost equally interesting, especially one in which the up-cast tubes connecting the upper and the lower drums being like the letter **U,** projecting horizontally, as in the case of the Du Temple boiler, with the fire underneath them. In another ingenious design by the same gentleman there was a similar boiler, but with one nest of larger tubes, the bottom parts of the nest forming the fire-grate; while in yet another instance another effective arrangement was attained by having two sets of tubes (right and left-handed) on this system, and interlaced into one another

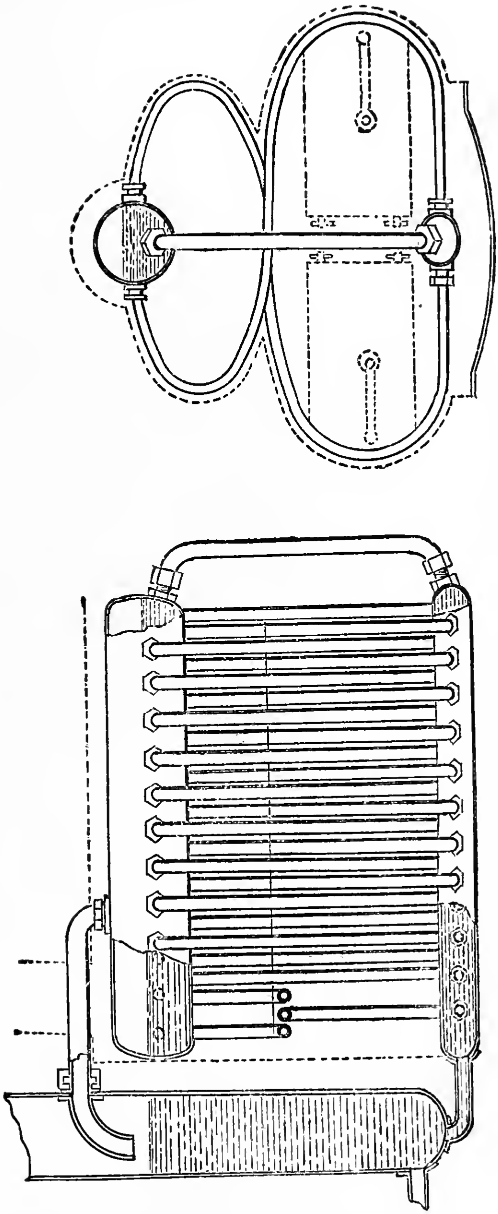


Fig. 215.—Goldsworty-Gurhney's Boiler, 1827.

so as to form a central furnace, the fire-bars, of course, being the lower part of the tubes. Gurney's boiler was intended for a working pressure of 130 lbs. per square inch; but even in those days it was anticipated that steam of much higher pressure might be used, and it was probably only the want of means of manufacture and the absence of a suitable design of engines for such pressures, as well as the necessity for obtaining chemically pure water, that prevented the general use of the tubulous boiler at a very early date.

From the beginning of the nineteenth century engineers have, both in this country and America, tried the water-tube type of boiler for high-pressure steam. That made by Col. Stevens in 1805 consisted of copper tubes 1 inch diameter and about 4 feet long. From his time to the present the number of inventions of a tubulous form of boiler is almost legion, as every engineer at some time of his life has experienced the necessity to satisfy one or other of the conditions that go to make the tubulous boiler superior to every other form. These conditions may be laid down as—

(1) Simplicity of form of element.

(2) Simplicity of construction due to smallness and lightness of element.

(3) Great strength of element in proportion to the working pressure and consequent large margin of safety.

(4) Small quantity of water contained, especially in proportion to the water evaporated per hour.

(5) General lightness of structure compared with the ordinary marine boiler.

(6) Immunity from injury, and eventual destruction by (*a*) rapidity of raising steam, (*b*) forcing when at work, and (*c*) sudden cooling by drawing or putting out fires.

Every water-tube boiler worthy of notice must fulfil the above requirements; but, to be successful for every-day use, it should also satisfy the following conditions:—

(1) The elements of the boiler must be so disposed and distributed as to be capable of taking up the heat generated by the fire without allowing to pass to the chimney more than is sufficient for draught purposes.

(2) The circulation of the water must be positive, continuous, and uniform.

(3) The internal and external parts must be capable of easy examination and cleaning.

(4) Every part liable to deterioration or derangement must be capable of removal with a minimum disturbance of other parts, or be put out of action without loss of general efficiency.

(5) The steam in the receiver must be properly separated so as to be fairly dry.

The generic form of a large class of water-tube boilers is pretty much that shown on Mr. Gurney's design (fig. 215)—namely, an upper chamber, into which the steam is delivered by a series of tubes surrounding the fire, and a lower chamber or chambers connected to this upper chamber by them and by downcast tubes, the latter serving to keep up the circulation through the other tubes.

Another favourite type, however, consists of horizontal, or nearly horizontal, tubes above and around the fire connecting two chambers, on the top of one of which is the steam drum, from which a connection is made to the other, to the bottom by preference, for proper circulation.

The great attraction for this latter type of boiler is the ease with which the tubes may be examined, cleaned, or removed, and the freedom from priming when the parts are properly proportioned and arranged. It has, however, the objectionable feature of flat surfaces which, in most of the forms, require staying.

A third type consists wholly of small pipes, the steam drum (if existing at all) being in a rudimentary form. It need hardly be said that this class of boiler is the lightest, but it is likewise the most dangerous of the water-tube type, and its employment is dependent on a regular supply of perfectly pure water. It is generally known as the flash boiler, from the fact that the water flashes into steam when forced into it.

The importance of the tubulous form of boiler has increased very much of late years, in consequence of the demand for extremely light machinery in the very high-speed vessels now required for war purposes, and the possibility of employing them successfully, is the result of the introduction of the evaporator, by which pure water can be distilled from sea water. For, although previously the surface condenser and a very small supply of fresh water carried in tanks permitted of the use of water-tube boilers in the mercantile marine, they were in almost every case a failure, owing to salt water eventually being used to make up the inevitable losses experienced in every ship; but the failure in these cases was mostly due to imperfect circulation. It is a little singular that the evaporator suggested and fitted by Hall to ships first 85 years ago having his surface condenser should have been allowed to go out of use and be forgotten.

The general adoption of the triple-compound engine was undoubtedly delayed for many years in consequence of the failure of the water-tube boiler in the s.s. "Propontis." But it must not be forgotten that, in past days, here and there water-tube boilers have worked successfuly under the ordinary every-day circumstances of a merchant ship including the use of salt water, and likewise that the boilers now fitted to ships have better means for examination and cleaning than existed in former times; besides which the supply of fresh water is practically ensured.

It is not claimed for these boilers that they are any more efficient as steam-generators than the ordinary marine tank boiler; but, on the other hand, it is not admitted that they are of necessity worse generators. On the contrary, it is very possible that, under centain conditions, a water-tube boiler may be more economical in steam production than the ordinary cylindrical boiler, inasmuch as it is capable of obtaining a better circulation with less expenditure of energy, and is not so liable to losses by radiation. Its efficiency, however, depends more on clean surfaces and tight casings than does the tank boiler, and generally the latter is easier cleaned, etc., than the former.

In marine practice, the heating surface in proportion to the grate has been generally small, and the makers and users of those boilers have been content with the other and real advantages derived from them rather than to effect the utmost economy by arresting the waste of heat which can, undoubtedly, take place in some forms. That the whole surface of the tubes is not efficient heating-surface goes without saying, just as no one counts as heating-surface the lower part of a furnace or flue, or the parts of a Lancashire boiler not exposed to heat. Probably only 0·7 of the circumference of the horizontal tube of these boilers acts as real heating-surface;

but, for convenience of expression, the whole tube hitherto has been counted. to serve frequently as an argument to their detriment by hostile critics.

It must also not be overlooked that, in some designs of water-tube boilers, a considerable number of the tubes, whose surface nominally counts as heating-surface, are practically not exposed to heat at all, and consequently really act as downcasts. Reckoning, however, the whole surface of all the tubes which are or may be exposed to heat as heating-surface, the allowance should be 40 per cent. more per I.H.P. than given usually for cylindrical boilers when weight is not of first consideration. In express boilers, where weight is of first consideration, as low as 1·5 square feet of nominal heating-surface per S.H.P. has given very fair results with a coal consumption of 75 lbs. per square foot of grate. It is, however, better to allow 1·8 square feet, and limit the consumption to 65 or 70 lbs., then 26 to 29 S.H.P. per foot of grate is obtained, and 1·15 lbs. of Welsh coal* is burned per hour for each square foot of heating-surface. That is to say, at or about 200 lbs. pressure an efficient water-tube boiler can evaporate 10 to 12 lbs. of water per square foot of nominal heating-surface per hour. Superheaters can be, and often are, fitted to water-tube boilers with advantage, and are a means of considerable improvement in efficiency; Mr. Harold Yarrow's experiments proving that, with a superheat of 100° F., his boilers gained in economy 8 to 10 per cent., and, with an increase to 150° F., the gain was even 11 to 13 per cent. Under these circumstances, when using 1·237 lbs. of oil fuel per square foot of heating-surface, as much as 14·6 lbs. of water were evaporated per lb. of fuel, and with a consumption of 0·542 lb. it was 15·9, or 8·6 lbs. per square foot of heating-surface.

Grate Area.—The area of the grate must in this type of boiler, as in every other, depend on the draught available, and consequently the size of grate may, to some extent, follow the ordinary rules given elsewhere; but in most forms of water-tube boilers it is not possible to work economically with so high a pressure in the ashpit as is used with the locomotive or ordinary marine boiler. In some few forms, the obstruction to the passage of hot gases is sufficiently great to permit of a higher pressure, but in almost every case a large grate is necessary for the water-tube boiler, in order to distribute the heat evenly over the whole of the boiler. Moreover, it is not a good thing to have the fire too intense when any portion of the tubes is touching it or near it; in fact, the tubes which are contiguous to the fire, and receive the direct radiation from the glowing coals, should be made of somewhat thicker material as well as larger in diameter. With express boilers, and the tubes arranged nearly vertical for rapid and certain circulation, the forcing of the fires may be made by an air pressure as high as 4 inches, and $3\frac{1}{2}$ to 4 inches is now the practice for full speed when the grates are getting dirty. The consumption of coal in such cases is 65 to 70 lbs. per square foot of grate, and 2·4 to 2·6 lbs. per I.H.P. (including that required for steering gear and all auxiliaries). 26 to 29 I.H.P. per square foot of grate is obtained. For turbines the consumption is not so large in normal conditions, being 1·65 to 1·8 per S.H.P. of coal, or 1·25 to 1·4 of oil.

Tubes.—It is well to arrange the tubes so that access to the more remote ones permits of their receiving a fair amount of heat, and prevents those nearer the fire from getting an undue amount. The tubes should be made

* If oil is burned the rate is 0·9 lb. per square foot of S.H.P.

of solid drawn steel, of best quality. Copper has been tried, but the difficulty of detection of a fault in solid drawn copper detracts very much from its usefulness, and is more than a set-off for the superior conductivity and the low corrosive character of that metal irrespective of the cost. The Admiralty insist on steel tubes being galvanised on the outside and not on the inside, the decomposition of the zinc when inside the tubes tending to produce hydrogen gas, by which an explosion might be caused on the incautious introduction of a lamp when empty. No doubt, for ships liable to be laying up out of use for long periods, the galvanising is an advantage; but for a ship that is generally in use, the necessity for it does not exist. There is, however, a further advantage in galvanising, inasmuch as in the process of pickling, defects can be discovered in the tubes which might otherwise escape detection; scouring with a fast-revolving brush will also answer this purpose. The small light boilers, employed in torpedo boats and torpedo boat destroyers, have up-cast tubes generally at least an inch external diameter and from 14 to 16 L.S.G. thick. They are expanded into the tube plates in the usual way, and when stay tubes are not used, the ends are slightly drifted so as to prevent drawing; if the tubes are galvanised, it is better to remove the zinc from the part fitting into the tube plate. For general purposes an inch tube is too small, except in small craft; in ordinary ships, as in the larger vessels for naval service, the tubes should be $1\frac{1}{4}$ to $1\frac{1}{2}$ inches diameter in certain forms of boilers, such as the Thornycroft, Yarrow, or Normand, and from $1\frac{1}{2}$ to 3 inches in those boilers in which the tubes are horizontal, or nearly so. Where weight is not of such paramount importance, the tubes of the horizontal boiler may be even somewhat larger still, especially those in the immediate vicinity of the fire.

Circulating Tubes.—The down-cast pipes are better to be of comparatively large size and few in number, so situated as not to be exposed to heat sufficient to convert them into upcasts,* and to be in the way of the natural horizontal flow of the water current in the steam receiver. These down-cast pipes are, as a rule, 3 to 6 inches diameter, but when a water wall is formed by tubes placed touching one another in the main part of their length, but their ends worked zig-zag fashion at the upper and lower receivers, they may be of the same size as the ordinary tubes, these, of course, acting as down-casts.

Stays.—In designing a water-tube boiler the same care must be exercised, as with the ordinary boiler, in arranging the stays so as to resist deformation and destruction. Many engineers, following the example of their locomotive brethren, depend on the ordinary tubes to effect this purpose; and, no doubt, so long as they are in normal condition they may be relied upon; but when the boiler is such that the tube ends might at any time become abnormally hot and be suddenly cooled, it would not be safe to depend on ordinary tubes for holding the parts together. When straight tubes are employed, stay tubes may, of course, be fitted, as in the ordinary boiler. General stiffness, as well as local stiffness, is usually obtained by making the tube plates abnormally thick, and, when the tube plate forms the part of a cylindrical drum, this thickness is necessary for strength to resist tangential strains.

* Mr. Yarrow's experiments show that the down-cast pipes may be heated to advantage under certain conditions, but it is certain that the continuous flow must not be jeopardised.

Steam Drums.—The size of the steam receiver is dependent, as usual, on the volume of steam produced, and in order to keep the size as small as possible, as well as to prevent priming, it is customary to work these boilers at a very high pressure, there being little difference in weight between such a boiler and one made for a much lower pressure. If, however, the boiler is to be worked at the same pressure as the engines, and that pressure is only 180 lbs., the steam dome must be of considerable size; in fact, it must follow the rules laid down for ordinary boilers; it is better, however, to use water-tube boilers with higher pressures, 200 lbs. being about the lowest advisable.

The Two Types.—The modern marine water-tube boilers may be roughly divided into two classes, the one having large or comparatively large tubes and suitable for large ships and long runs, the other having small tubes and suitable for small ships or for short runs at very high speed, and consequently called "Express Boilers." Each of these can be subdivided into two divisions, the one having drowned tubes (that is, tubes whose upper ends are submerged in water), and the other having the upper ends exposed to steam only when the circulation is maintained by priming or raising of the water with the steam in the pipes, and the discharging of it into the bottom of the steam receiver. It is claimed by the makers of the latter class that the steam obtained is quite as dry as that produced by the submerged tube boiler. Of course, dash plates and other means are provided for separating the steam from the water besides the usual internal steam pipe: even with submerged tubes it is sometimes found advantageous to fit baffle plates so as to ensure steady circulation and prevent priming. The design of the priming boiler is one that admits of a greater length of tube and a longer exposure to heat than is usual with the other forms in which the tubes are straight or nearly straight, and their advocates claim for them an elasticity and power of resisting sudden changes of temperature superior to that of the straight tube boiler. On the other hand, they have the obvious disadvantage of cost of manufacture as well as impossibility of internal examination, besides the difficulty and costliness of removing and replacing an injured tube.

Herreshoff Boiler.—In the very early days of the last century others, besides Mr. Jacob Perkins, attempted to produce high-pressure steam with the lightest possible apparatus, by making a boiler entirely of small tubes, and another inventor went so far as to construct one consisting of a single tube coiled roughly into the shape of a bell or beehive, and placed this over a fire. Water was pumped into the lower end of this coil, and steam emitted from the upper end. As might be anticipated, the generation of steam was most rapid, and attended with considerable risk to the onlookers. So long as the feed supply was regular and the water perfectly pure, this boiler could be used with impunity; but even now, as in those early days, no feed pump could be relied upon to work with the necessary uniformity, and as in practice, all water contains more or less solid matter, such boilers were found impracticable, and the more so as, from the curved form of the tube, it was impossible to examine or clean them mechanically. The Herreshoff boiler is a modern form of this type, which has always been a favourite one for certain purposes in America; its inventor subsequently perfected a design of it which worked with a considerable degree of success, and it was introduced by him into this country, at the time that Mr. (now Sir J. I.) Thornycroft introduced a similar boiler as regards coils, and fitted one in the s.s. "Peace" (1883). Eventually, however, it met with the fate of its prototype, so that it is now seldom or never used. His boiler consisted of

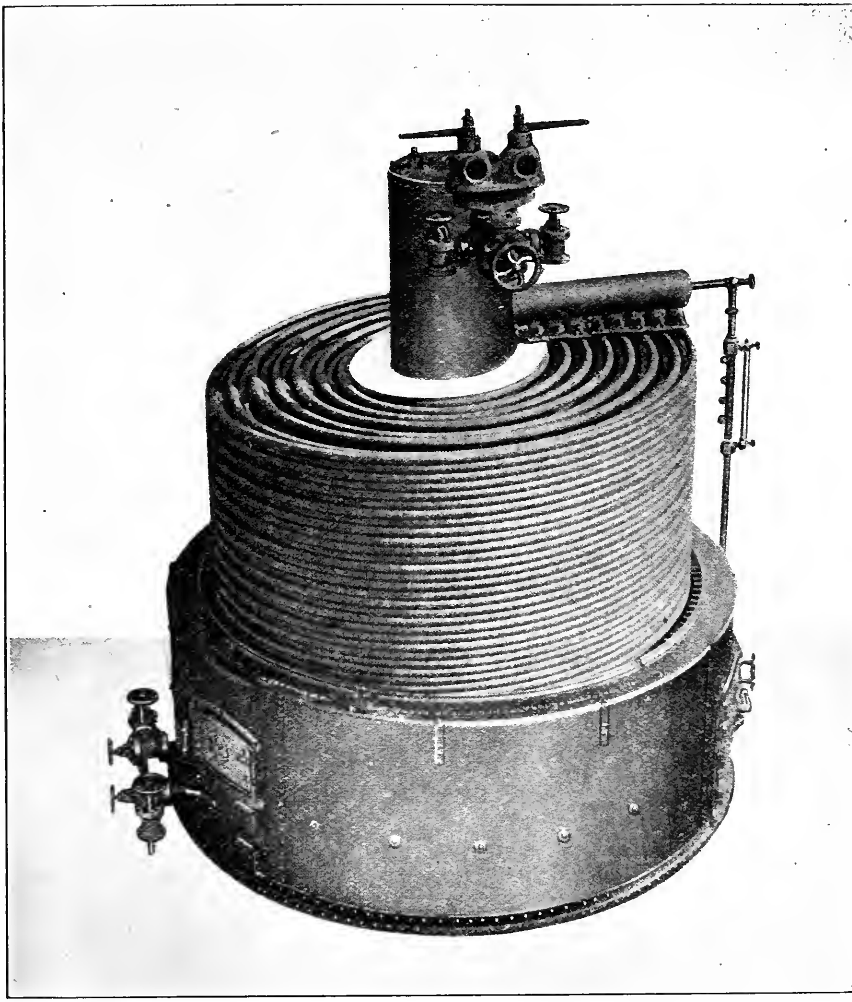


Fig. 216.—Ward's Sectional Coil Boiler—Heating Surface, 2,490 square feet; Grate, 77 square feet.

two coils of pipe; the outer, which is in the form of a closed cylinder, is made of tubes of uniform section, connected at their ends by welding so as to be continuous; the inner, which is in the shape of a bell, is likewise made

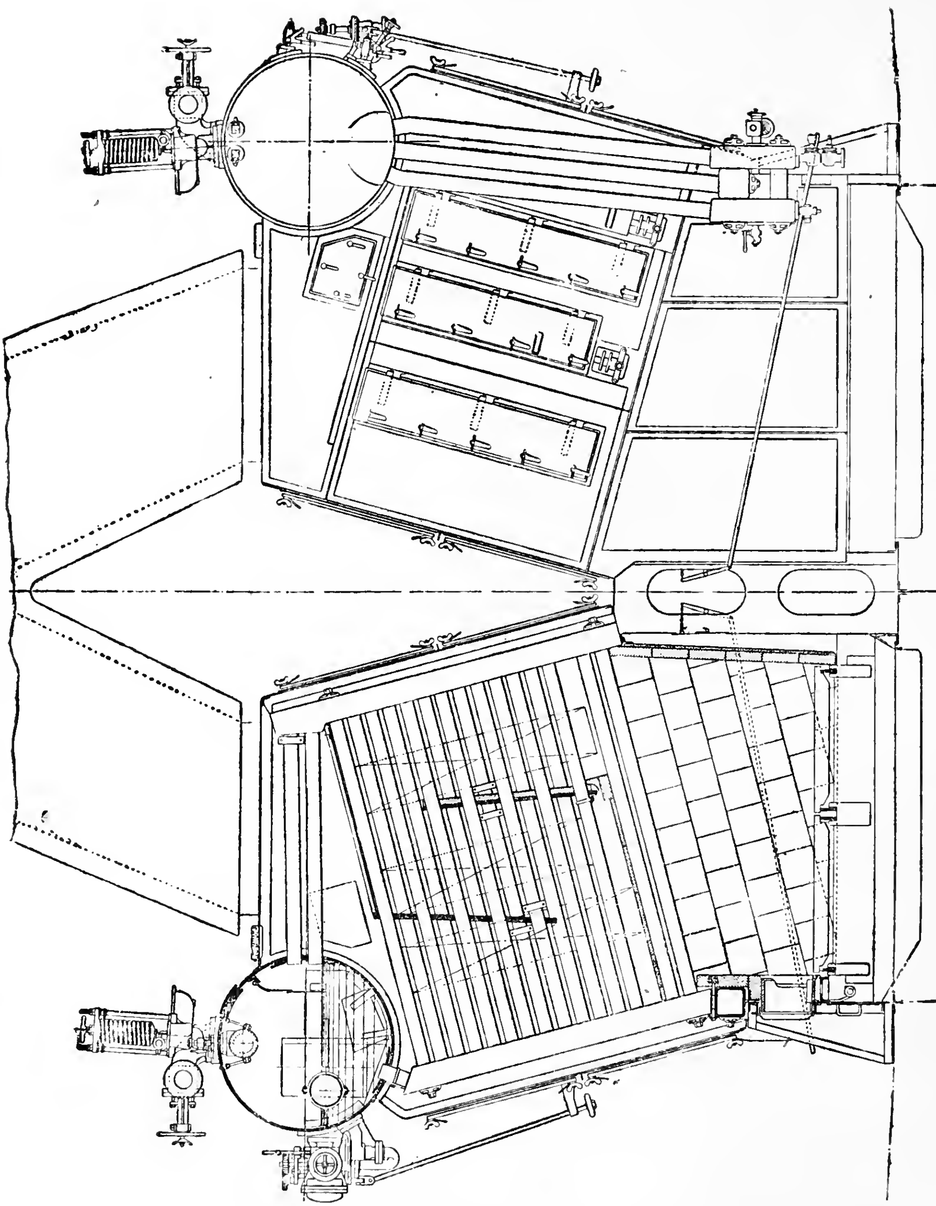


Fig. 217.—Babcock & Wilcox Naval Boiler (Large Tube).

of continuous tubes, but those forming the sides of the bell are of larger section than that forming the crown. The top of the outer coil is connected to the top of the inner, the bottom of the outer is connected to the feed pump, and the bottom of the inner to a steam drum or receiver, from which the engine takes steam. The fire is in the inside of the bell, which is supported on brickwork, and is similar to that of a vertical donkey boiler. The outer coil is cased with sheet iron, and the inner is partly so covered. The water, in traversing the outer coil, is gently heated by the waste heat; on traversing the crown of the bell it is rapidly heated, so that when it enters the larger pipe it is in a state of ebullition, part steam and part water; when it has traversed the whole length of the coil it should be all steam. Another coil boiler, well-known in America, is Ward's; it has been used in the U.S. Navy, and of its kind is a good one (*v.* fig. 216). It, as well as Thornycroft's and Herreshoff's, has, however, the fatal fault of inaccessibility for cleaning, but with anthracitic coal giving off little or no smoke or tarry vapour, and using pure water, this fault is not of serious consequence.

Babcock & Wilcox Boiler (fig. 217).—The two gentlemen whose names were joined in the patent and gave the title to the Company that manufactures these boilers, devoted many years to the perfecting of their system. In this case the flat boxes of the Watt and D'Allest boilers are replaced with narrow sinuous chambers (called "headers," fig. 218), placed nearly vertically side by side. With this arrangement staying is unnecessary, but opposite the tube ends are the usual openings, as in the above-named boilers, fitted

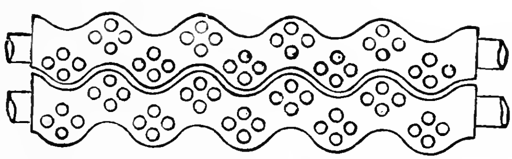


Fig. 218.—Headers for Small Tubes.

with doors or plugs. The top of each front header is connected by a pipe to a drum of considerable size, placed immediately over the back headers, and to which it is connected also by pipes. The front and back headers are connected by a large number of inclined tubes; but in the case of this boiler they are of much smaller diameter than in others of the same type. The circulation of the water is obvious and good, and of such a nature that this boiler may be classed as a priming one. The furnace is, of course, beneath the tubes, and extends nearly to the full width of the boiler. Sometimes on each side of the boiler, forming a water wall, is a system of small vertical tubes, fitted into a horizontal box at the bottom, and into a similar horizontal box of square section at the top. The top boxes are connected also to the steam drum, and at the opposite ends the top and bottom boxes are connected by external circulating tubes. These boxes and the "headers" are of square section, and made of wrought iron or steel, welded up and neatly finished. The tubes throughout are simply expanded in the tube holes, and although it appears to be a most difficult operation to thus secure them, and a still more difficult one to tighten them if they leak, in practice all such difficulties have been overcome, and these boilers found to work most satisfactorily. This, however, is now seldom supplied, as brickwork to the height of the lower tubes is better, and a good casing above in wake

TABLE LVI.—Basin and Sea-going Trials of H.M.S. "Sheldrake" in 1898-99 (Babcock & Wilcox).

Nature of Trial.	No. of Boilers in use.	No. of Boilers in reserve.	Duration in hours.	Steam Pressure.	I.H.P.	Air-pressure in Stokehole.	Coal per I.H.P., all purposes.	Coal per Sq. ft. Grate.	Water per Pound Coal.	Ashes per Pound Coal.	Grate Surface.
Evaporative, ...	2	2	8	168	1116	·0	1·69	15·0	10·5	...	126
"	2	2	8	179	1292	·0	1·46	15·0	10 5	...	126
"	2	2	8	169	1761	...	1·78	25·0	9·05	...	126
"	2	2	8	165	1873	...	1·67	25·0	9·34	...	126
8 hours at 2500 I.H.P., .	4	...	8	152	2642	·0	1·43	15·0	...	...	252
3 hours at 3000 I.H.P., .	4	...	3	151	4050	·43	1·57	25·6	...	...	252
3 hours commissioning, .	4	...	3	119	2735	·14	1·64	17·8	...	...	252
1000 miles at 1500 I.H.P.,	3	1	69	120	1303	·0	1·61	12·8	...	·152	189
" " 1500 "	3	1	68	120	1506	·0	1·6	12·67	...	·150	189
" " 1500 "	3	1	70	135	1534	·0	1·75	14·2	...	·200	189
" " 1500 "	3	1	68½	130	1539	·0	1·59	13·1	...	·150	189
" " 1800 "	3	1	67	135	1829	·0	1·6	15·4	...	·216	189
" " 1800 "	3	1	66½	140	1838	·0	1·68	16·4	...	·220	189
" " 2000 "	3	1	59	145	2033	·0	1·57	17·0	...	·220	189
" " 2000 "	3	1	61½	140	2042	·0	1·56	16·8	...	·230	189
" " 2250 "	3	1	56¾	150	2245	·0	1·63	19 4	...	·250	189
Total, ...	...	...	632¼	...	...	...	...	...	...	...	...
Mean, ...	...	...	...	146	1974	...	1·63	...	9·85	·198	...

of the tubes answers every purpose. It need hardly be said that the use of clean, soft water is essential. In the later marine type the tubes adjacent and near to the fires have been enlarged very considerably, so that, instead of being about 1½ inches diameter, as were the other tubes, they are now at least 2½ inches diameter (*v.* fig. 218*a*). These boilers are lighter in proportion to their output of steam than the ordinary cylindrical boiler; but they are not so light as some of the other forms of water-tube boilers; in

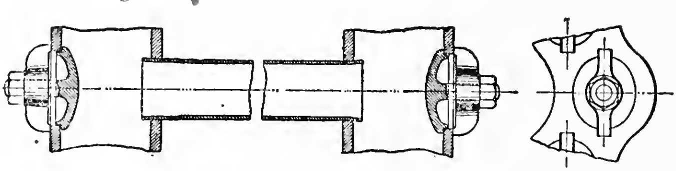


Fig. 218*a*.—Babcock Tube and Headers.

this case, however, there is not the same necessity as exists in others for working at a high pressure to be successful, and in this respect it is much superior to one or two of the other forms of water-tube boiler when required to be of large size. In the American Navy, as in H.M. service, these boilers have given great satisfaction, and proved themselves to be the most formidable competitors for general adoption, both against the cylindrical and the other types of large water-tube boiler. The simplicity of design and cheapness of manufacture will also recommend them for use generally. In H.M.S. "Sheldrake" these boilers, with best Welsh coal and a consumption

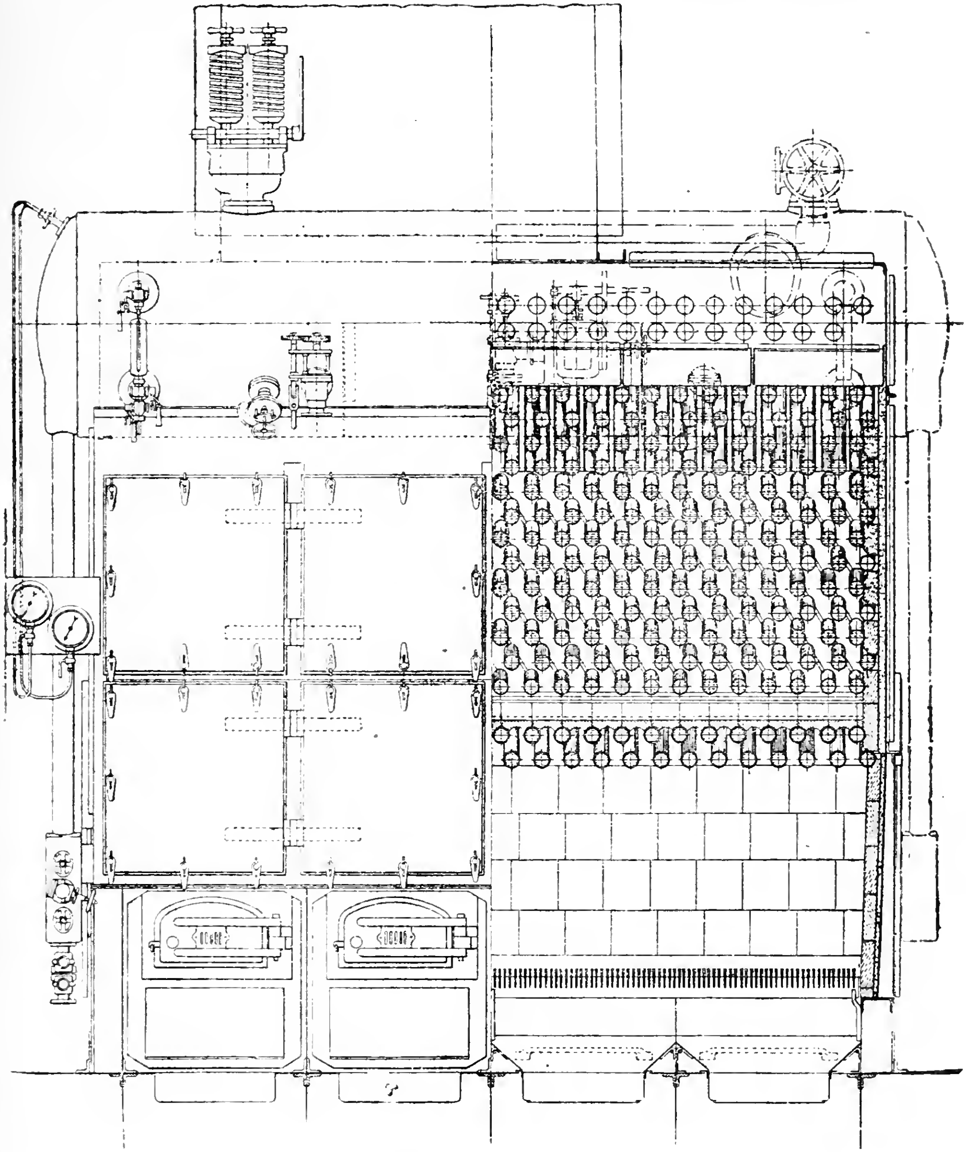


Fig. 219.—Babcock & Wilcox Boiler (Small Tube), Naval Design.

of 40·7 lbs. per square foot of grate, evaporated 9·15 lbs. water per lb. of coal, from and at 212° F., and at a half that rate the evaporation was 11 lbs., the

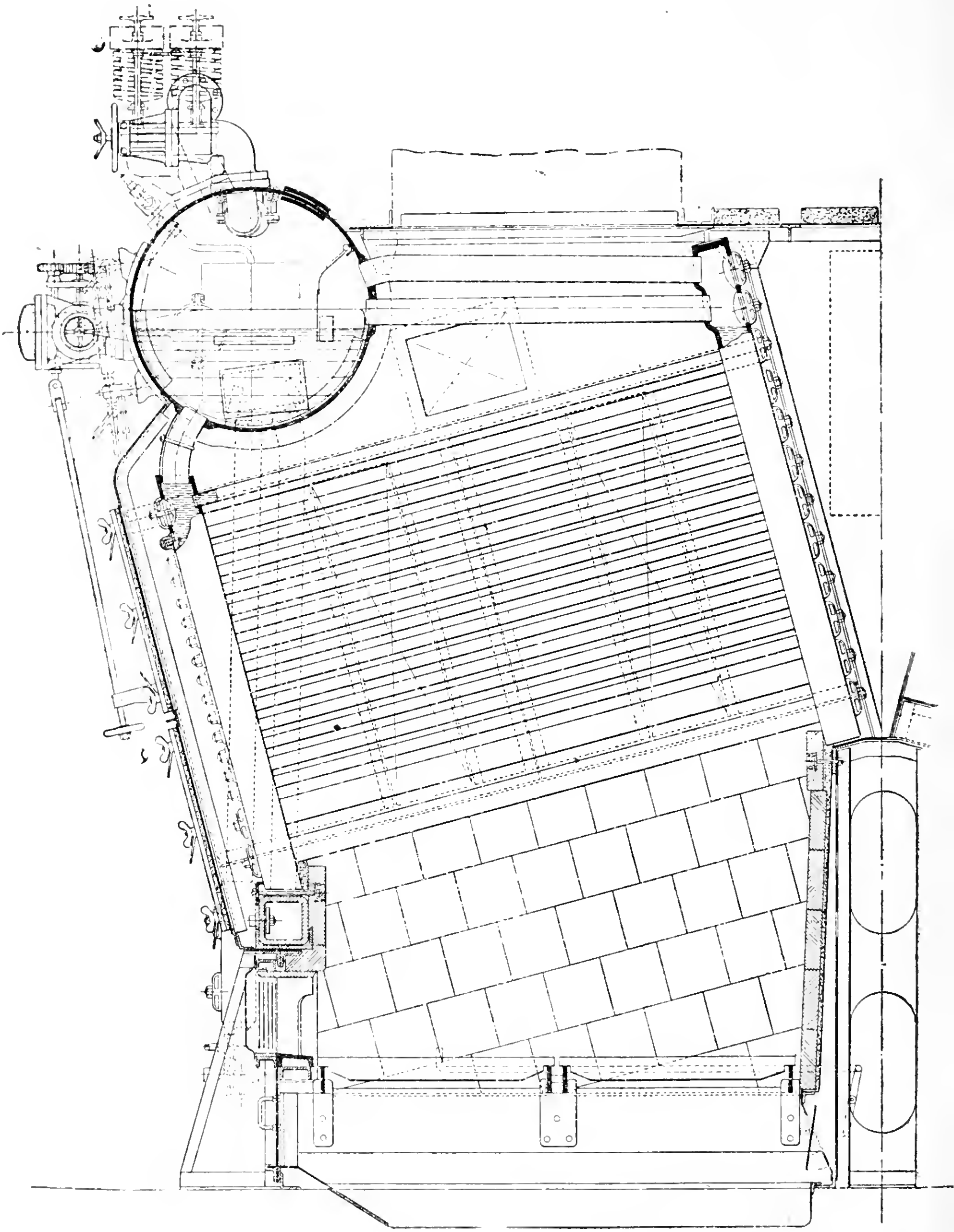


Fig. 220.—Babcock & Wilcox Naval Boiler (Mixed Tubes).

evaporation per square foot of heating surface being 8·3 and 5·0 lbs. respectively. At sea, and when running full speed, the consumption per I.H.P.

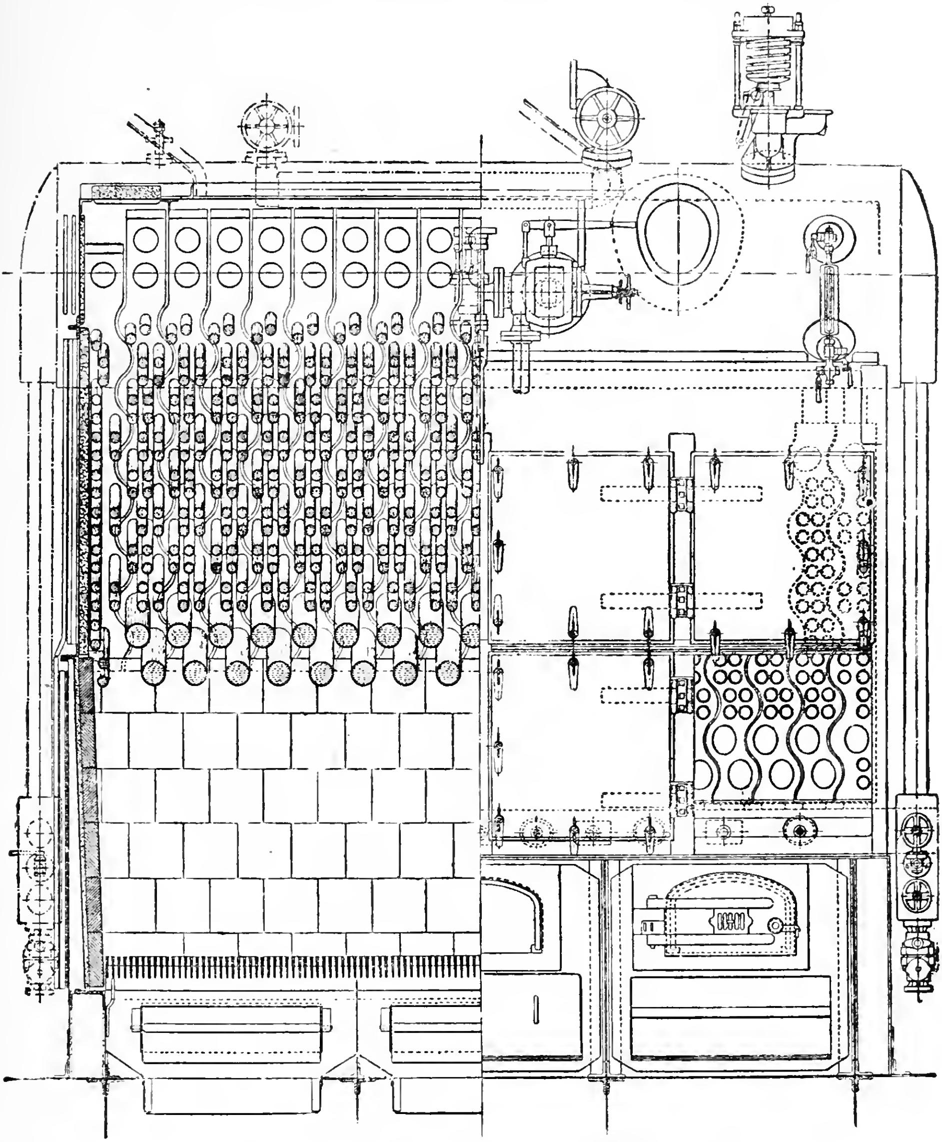


Fig. 221.—Babcock & Wilcox Naval Boiler (Mixed Tubes).

was only 1·57 lbs., and at two-thirds power 1·429 lbs. Equally good results have been obtained in U.S. Naval ships with Pocahontas coal.

The Babcock & Wilcox Company sometimes fit into the uptake what is virtually an extension of the boiler, and call it a feed heater; this, of course, materially adds to the economy of the system, as it also adds considerably to the weight.

The Special Boiler Committee appointed by Parliament reported favourably on this boiler, as one of those suitable for use in the Navy; consequently a large number of ships—mostly battleships and large cruisers—are fitted with them. The Admiralty require such boilers to evaporate 12 lbs. of water from and at 212° when the coal consumption is at the rate of 18 lbs. per square foot of grate, $11\frac{1}{2}$ lbs. when the rate is 24 lbs., and 11 lbs. when 30.

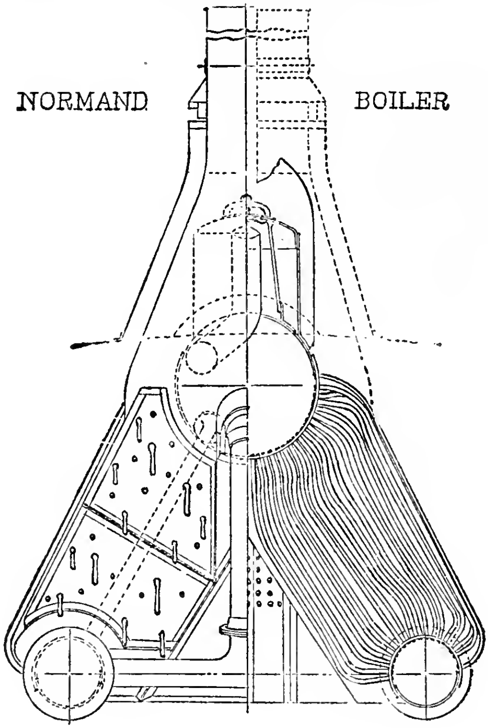


Fig. 222.—Normand Boiler.

Fig. 217 shows the naval boilers, which are made entirely of large tubes, and the baffling arrangements, which have been found beneficial on shore; this type is also the one best suited for the mercantile marine. Fig. 219 is a design of boiler made entirely of small tubes, suitable for cruisers and express steamers generally. Fig. 220 shows the boilers of the cruiser type, which are of a lighter design than fig. 217, having large tubes at the bottom only, and smaller higher up; no baffles are deemed necessary.

Normand Boiler.—Fig. 222 shows the form of boiler of the late M. Normand, of Havre. It consists essentially of one horizontal upper drum of considerable diameter, and two lower drums of much smaller diameter, the lower ones being placed on each side of the fire, and connected to the upper one by tubes of small diameter, bent to such form as to obtrude the greater part of their length into the space above the fire, and so receive heat from it, and at each end so as to be normal to the surface into which they fit. The upper and lower drums are likewise connected at their ends by larger tubes, which act as downcasts, so that the water circulates up through the smaller tubes into the upper chamber, delivers the steam with which it is charged, and flows away to the end of it, and down through the outer tubes to the water drums below. This boiler exposes a very large amount of surface, both to the direct action of the fire and to hot gases. The circulation is perfect; there are no flat surfaces requiring to be stayed; the drums themselves are of the cylindrical form, with spherical ends, so as to be as light as possible consistent with strength; and by the disposition of the tubes at the front and back there is a minimum amount of brickwork required for a boiler of this kind, while the tubes have a sufficient amount of curvature to permit of considerable elasticity in the whole structure, so as to enable it to withstand the rough usage of rapidly raising steam. On the other hand, the tubes cannot be cleaned mechanically nor examined, either internally or externally, and, in the event of a tube splitting, or becoming otherwise so damaged as to necessitate removal, unless it be an outside one, great expense is entailed by having to remove the other tubes that surround it to get at it. M. Normand arranged the inner and outer rows of tubes to form water walls on each side of the grate, and on each outside of the boiler: he left, however, some open work on each side of the grate near the front, through which the hot gases can pass among the tubes on each side to the back end of the boiler, where there are additional tubes massed behind the brickwork of the furnace. The hot gases then pass upward into an uptake leading to the funnel base. By this method the hot gases are made to flow uniformly over practically the whole of the tube surface. The Normand boilers in H.M.S. "Ferret" had 8,112 square feet of surface and 154 square feet of grate, and gave steam for 4,774 I.H.P. Similar boilers in the French ship "Forban" proved very economical, the consumption of fuel at full speed being only 1·36 lbs. per I.H.P. with triple reciprocators.

The Normand Siguady boiler consists of two Normand boilers placed back to back, and connected by the steam collectors and water drums.

Ferguson & Fleming's Boiler (fig. 223).—Messrs. Ferguson & Fleming make a boiler similar in principle to the original Normand, and get over the difficulty of drawing the tubes by limiting their length from end to end to the diameter of the upper drum, so that each and every one of them can be drawn into this drum without affecting the surrounding tubes.

Seaton's Boiler (fig. 224).—In this boiler there are two water drums at the bottom, as in the Normand, and two steam drums at the top; there is an intermediate water drum between them of sufficient size to admit of the tubes connecting it to the upper and lower drums being drawn into it and removed. In this case the tubes are straight, so that from the middle drum every tube can be examined and cleaned, as well as removed; the transverse section of the boiler is the shape of the letter X. In addition to the advantages of being able to examine, clean, or remove every tube at the middle chamber, there is in this boiler the additional one of the hot gases being forced to flow twice through the tube before emerging into the funnel. The upper and lower drums in this boiler are connected by a water wall of tubes, which act as downcasts.

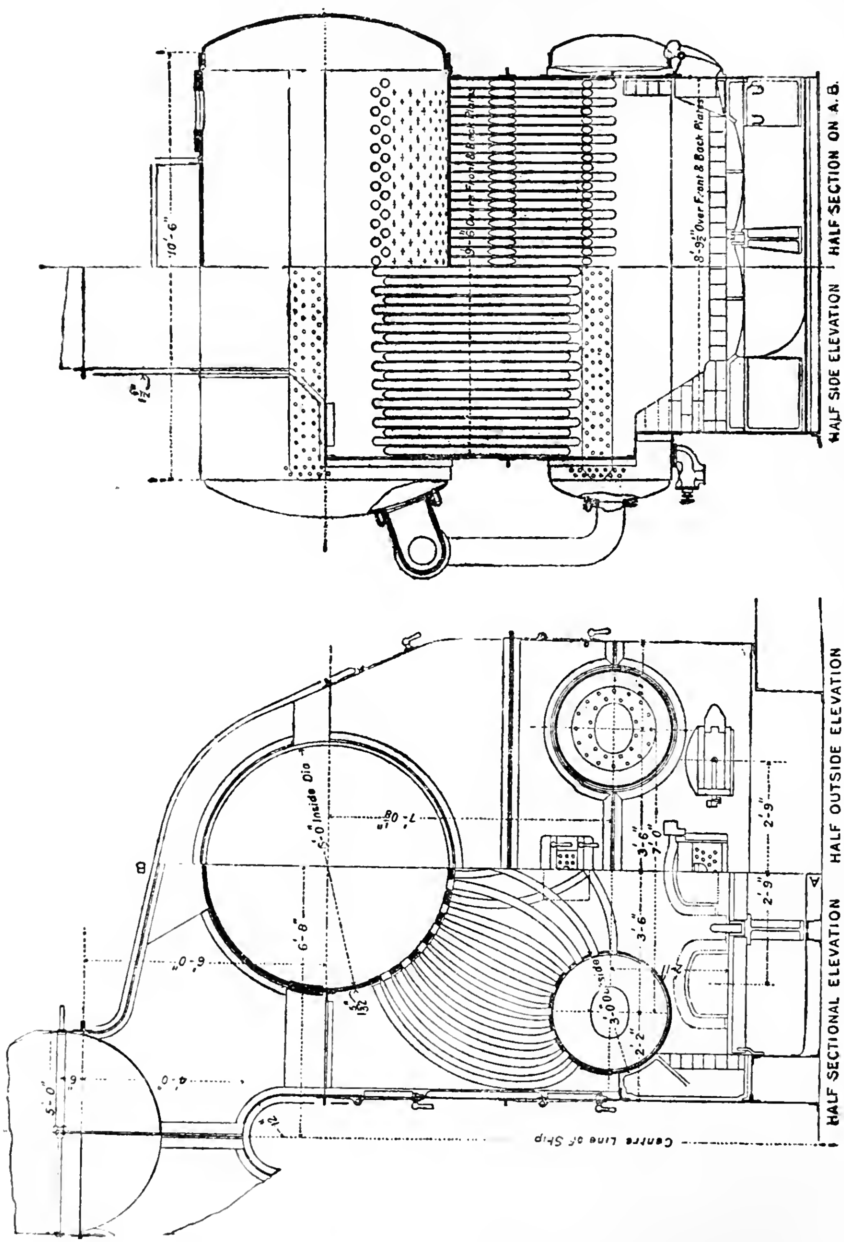


Fig. 223.—Fleming & Ferguson's Boiler (Marine Design).

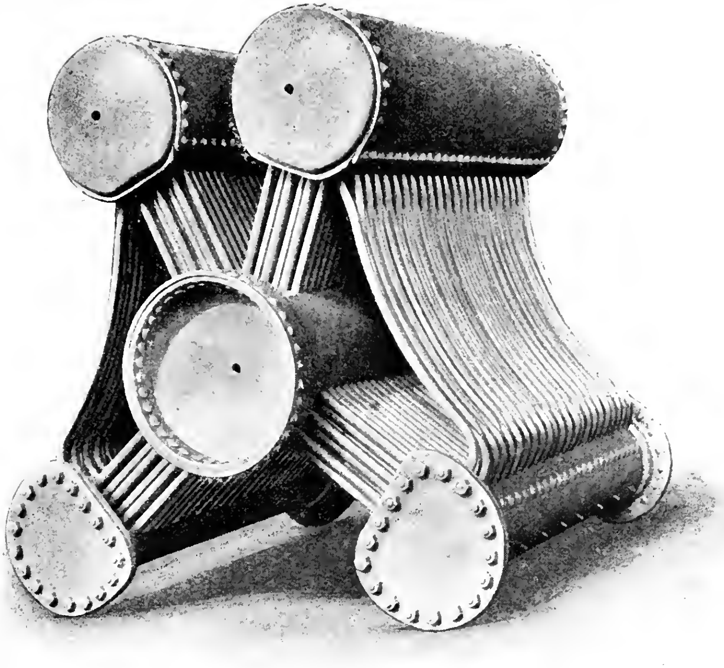


Fig. 224. - Seaton's Boiler.

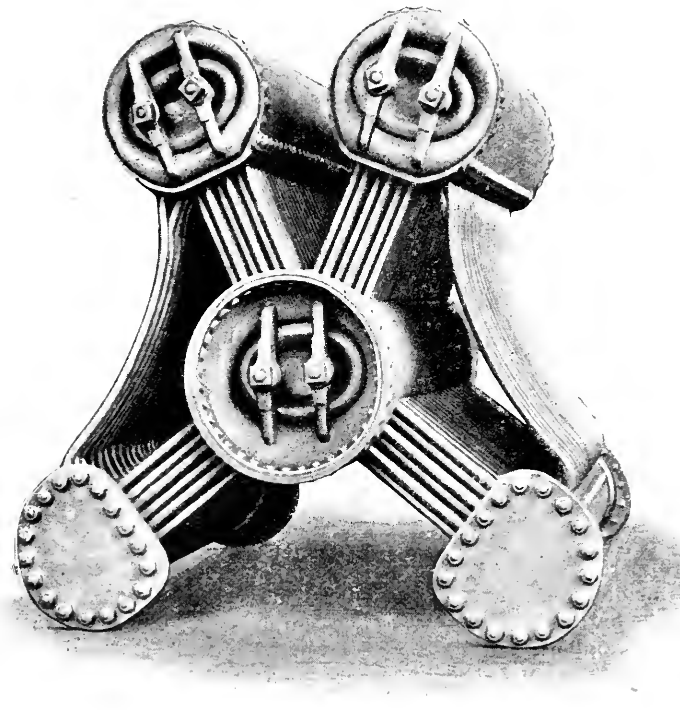


Fig. 224*a*.—Seaton's Boiler.

Yarrow Boiler (fig. 226).—This boiler, the invention of Mr A. F. Yarrow, has proved to be one of the most successful in practice, both in the production of steam, and its capability of being examined, repaired, or cleaned, and it can, besides, withstand the same rough usage as the Normand. It differs from the Normand boiler, however, in having straight tubes only, so that

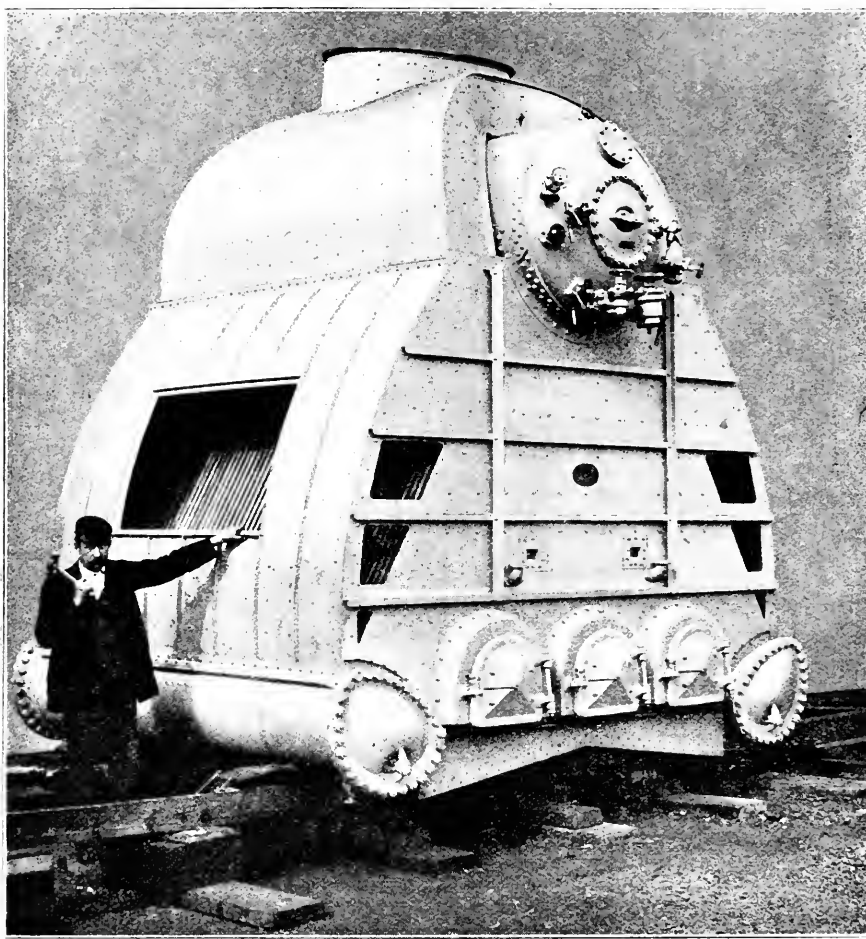


Fig. 225.—Yarrow Boiler (Large Tube Type).

from the upper drum they can all be quickly examined and cleaned. In the earlier forms of it, this drum was made in halves, the top of it being connected to the lower by a flange with bolts and nuts; it was, therefore, capable of easy removal, and when this was so the bottom of the boiler consisted, as in the Normand, of two cylindrical drums; but as in the larger boilers

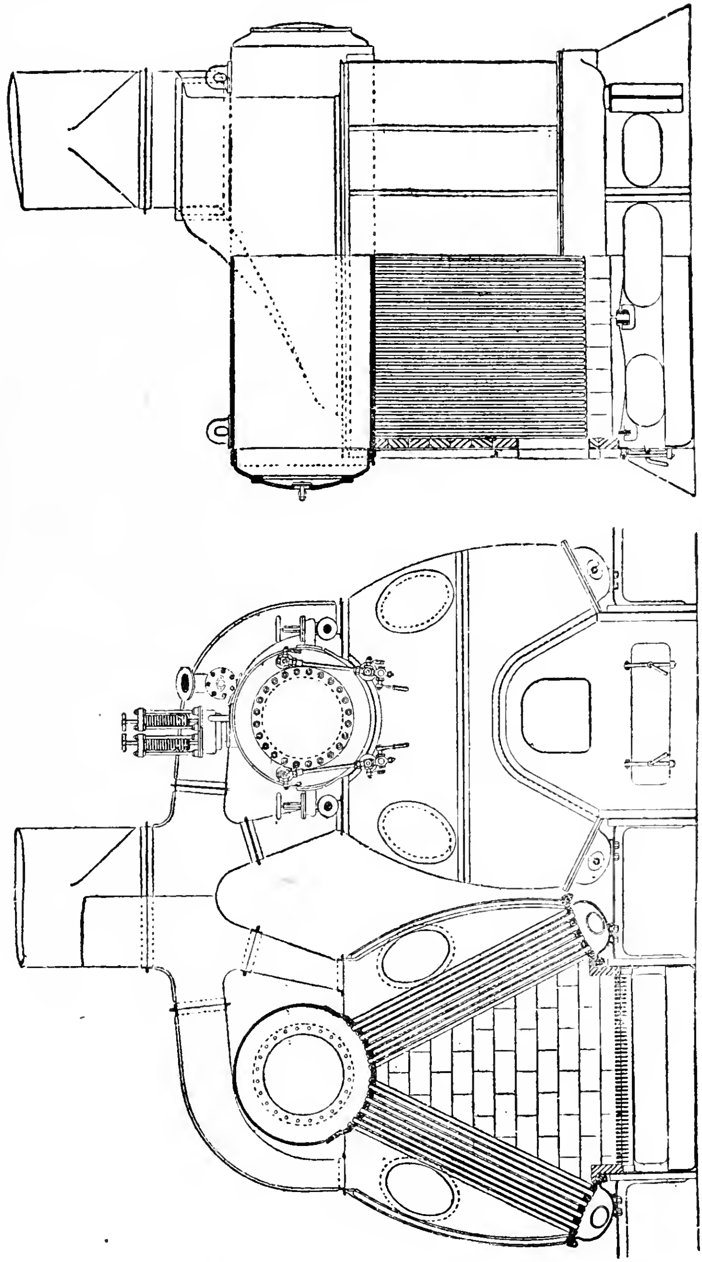


Fig. 226.— Yarrow Boiler

it was found almost impossible, as well as inconvenient, to have a bolted joint in the upper drum, the lower drums were made in halves, being formed with a tube plate and pocket bolted to it, as shown in the illustration, and to remove a tube necessitates the taking off of one or other of the pockets.

The Yarrow boiler was one of the few recommended by the Boiler Committee to the Admiralty for further trials and use in large ships, but with larger tubes than usual in *Express* boilers. This kind (fig. 225) has been fitted in many British and foreign warships to be used in connection with cylindrical boilers. The lower or water drums are in this case made approximately cylindrical, like those of M. Normand, and the tubes are removed and replaced by removing those in direct line with the ones requiring it, and dodging from hole to hole as far as the slackness of the new tube in the old holes will permit. The Yarrow boiler is now used very extensively in cruisers and battleships, as well as in the smaller craft; its straight tubes in a nearly vertical position render it a very trustworthy and efficient one. It has also

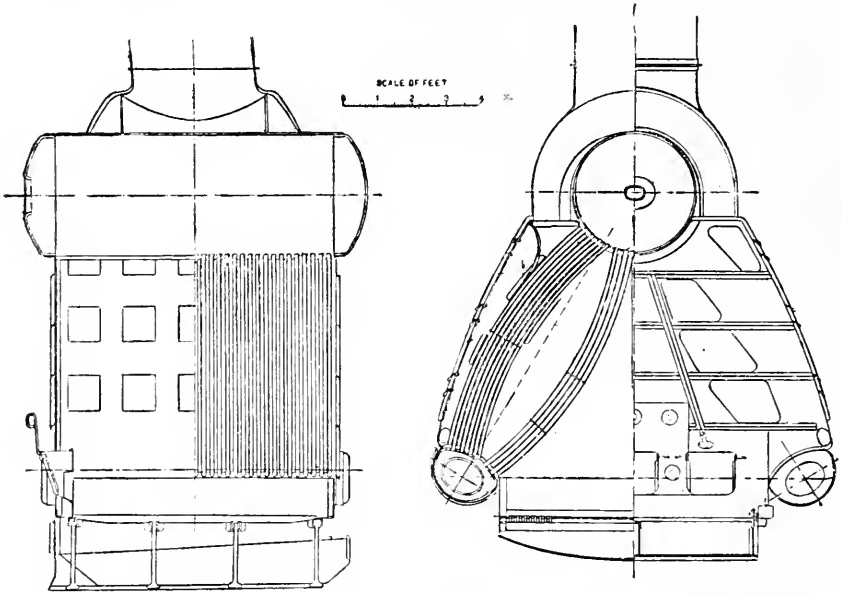


Fig. 227.—Japanese Navy Boiler (Yarrow Type) for Destroyers and Torpedo Boats.

been adopted in foreign Navies for cruisers and small craft. Fig. 227 is an example, being the particular form of it developed by the Japanese.

Blechynden's Boiler (fig. 228) is like the Yarrow as now made, but the tubes are very slightly curved, and the upper end enlarged so as to permit of them being withdrawn through a few small holes on either side of the top middle line of the steam drum; in recent boilers the tubes have been arranged to draw through one set of holes on this centre line. A double row of tubes forming water walls on each side, with the usual openings at the upper part, was often added to act as down-comers as well as walls.

White Forster's Boiler is somewhat like Blechynden's, but it has the tubes curved in a longitudinal plane so as to be drawn into the drum and through the manhole at its end.

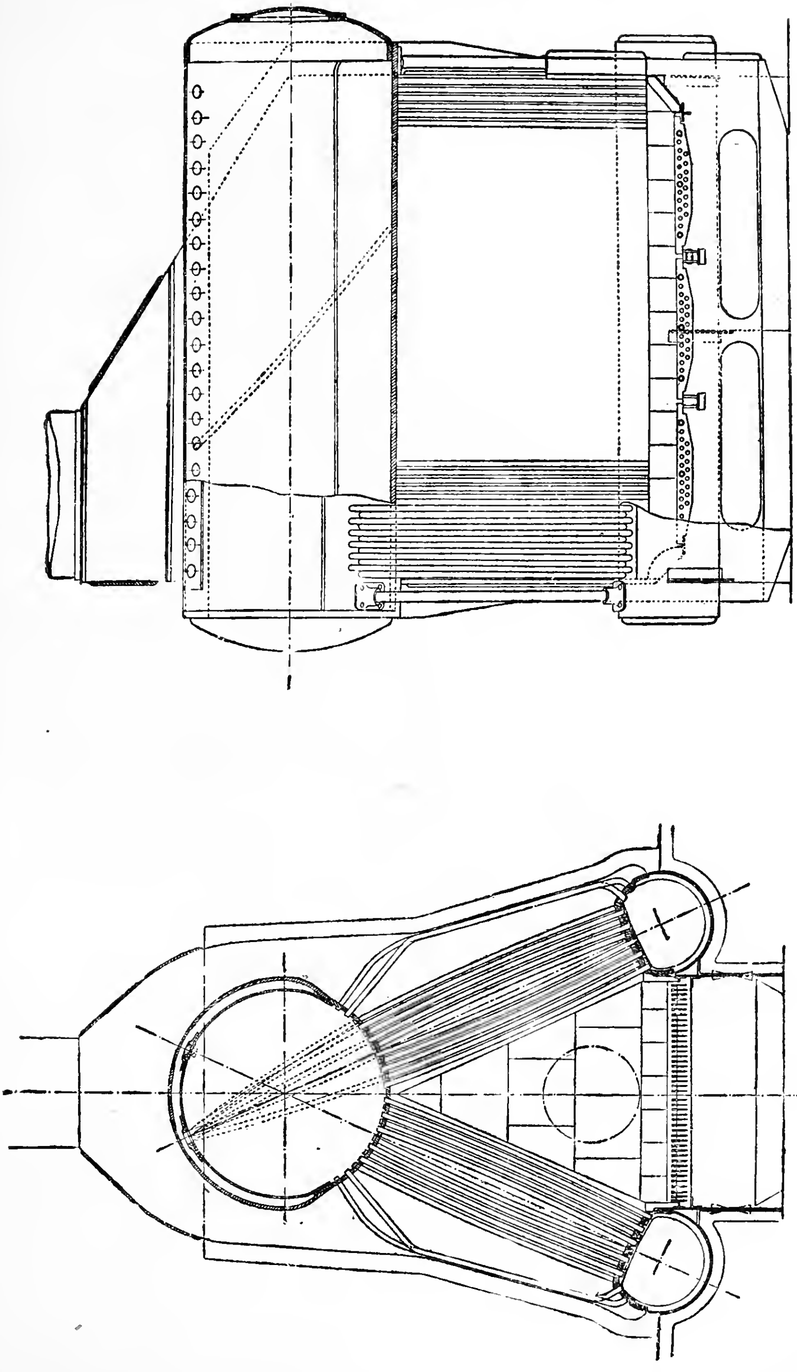


Fig. 228.—Blechynden's Express Boiler.

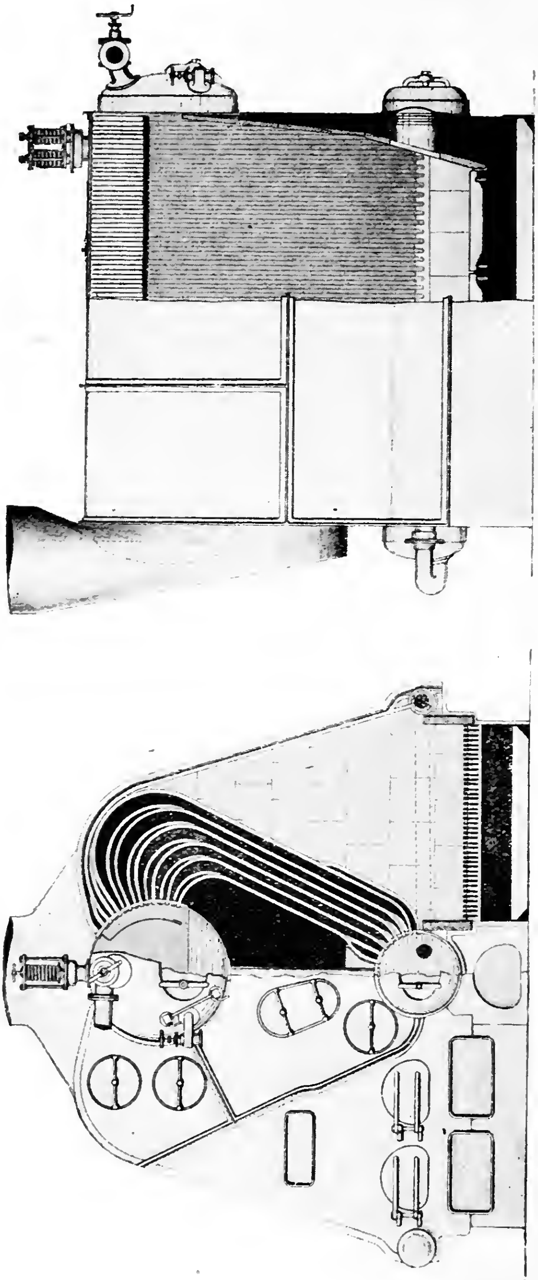


Fig. 229.—Thornycroft Boiler, the "Daring" Type.

Thornycroft's Boiler.—The boiler known as the "Speedy" type was the direct descendant of Goldsworthy-Gurney's invention of 1827 (see fig. 215), and consisted essentially of the top and bottom drums, as in the Normand and Yarrow types, but the tubes in this case were not only bent in so as to obtrude themselves on the space over the fire, but they were carried up and around so as to enter the top of the drum instead of the bottom; it is, therefore, distinctively a priming boiler, inasmuch as there can be no circulation without the flow of water through the tubes into the steam space. There were the usual downcast pipes at each end, which were of considerable size. In this boiler there is, of course, a greater length of tube exposed to the hot gases, and consequently, if necessary, a large amount of surface per foot of grate can be provided. It had considerably more elasticity than the Normand type of boiler, and likewise possessed the defects of the latter, as it was absolutely impossible to examine the tubes, and, in case of damage to one of them, the difficulty of removal and replacement considerable. Owing to the extreme curvature of the tubes, there is the greater liability of the tubes to choke with foreign matter that may be in the boiler, or get into it with the water. For these reasons it was objected to by engineers, and soon discarded by the Admiralty, although experience proved it to be a very efficient evaporator, a rapid generator of steam, and capable of very rough usage. Fig. 229 is a further invention of Sir J. I. Thornycroft, and known as the "Daring" type. With the same facility for getting sufficient heating surface as with the *Speedy*, this boiler permits of a larger grate, and allows of a better circulation of hot gases as well as better uptake and funnel; it has, however, the same objections due to the great curvature of its tubes.

Mumford's Boiler (fig. 230) is a modification of the Fleming & Ferguson (fig. 223), made to suit the service conditions in cruisers and small craft; it is very simple in design and construction, and although the curvature is too great to permit of a visual examination of the tubes, it enables them to resist the tendency to contortion from the severe heating some of them get when working under forced draught. Moreover, each one of them can be withdrawn into the steam drum, and from it passed out for careful examination in daylight, and likewise replaced in an equally easy and expeditious way. These boilers are found to be safe and economic in working of launches, pinnaces, and torpedo boats, and equally efficient when of the larger sizes now made by Mr. Mumford, where the total heating surface in a single-ended unit is 1,180 square feet, the tubes in that case being $1\frac{3}{4}$ inches diameter, 10 S.W.G. thick, and varying from 4·7 to 6·0 feet long; its steam drum has a diameter of 5 feet.

Du Temple Boiler.—This consists of the usual upper drum, which is of considerable size, and two lower chambers, of square section, placed above the fire level on each side of the grate. The top drum is connected to them by a nest of tubes bent zig-zag, so as to cross the flow of gases five times. The portion of tube exposed to direct action from the fire is generally made of smaller diameter than the rest of its length. There are the usual down-cast pipes at the end. This boiler possesses many of the characteristics of the Thornycroft boiler, including some of the evils in a magnified form. The contraction of diameter of the tubes in the neighbourhood of the fire has certain advantages; but, on the other hand, modern practice has shown that there should be larger tubes exposed to direct radiation from the fire,

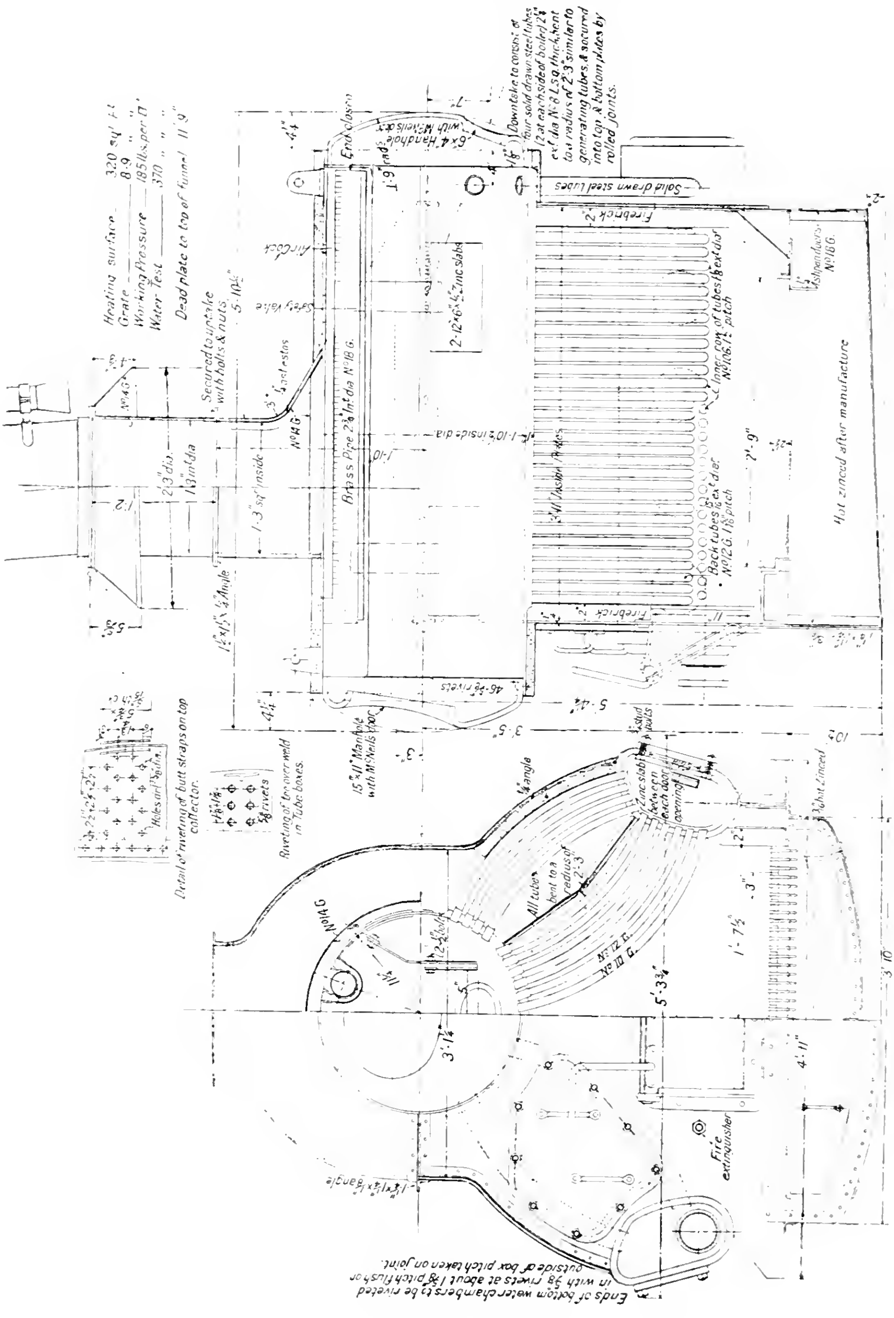


Fig. 230.—Mumford's Large Water-tube Boiler.

and that those tubes in the immediate vicinity, although not exposed to direct radiation, are the better for enlargement.

Reed's Boiler.—Fig. 231 is a view of the boiler designed and patented by

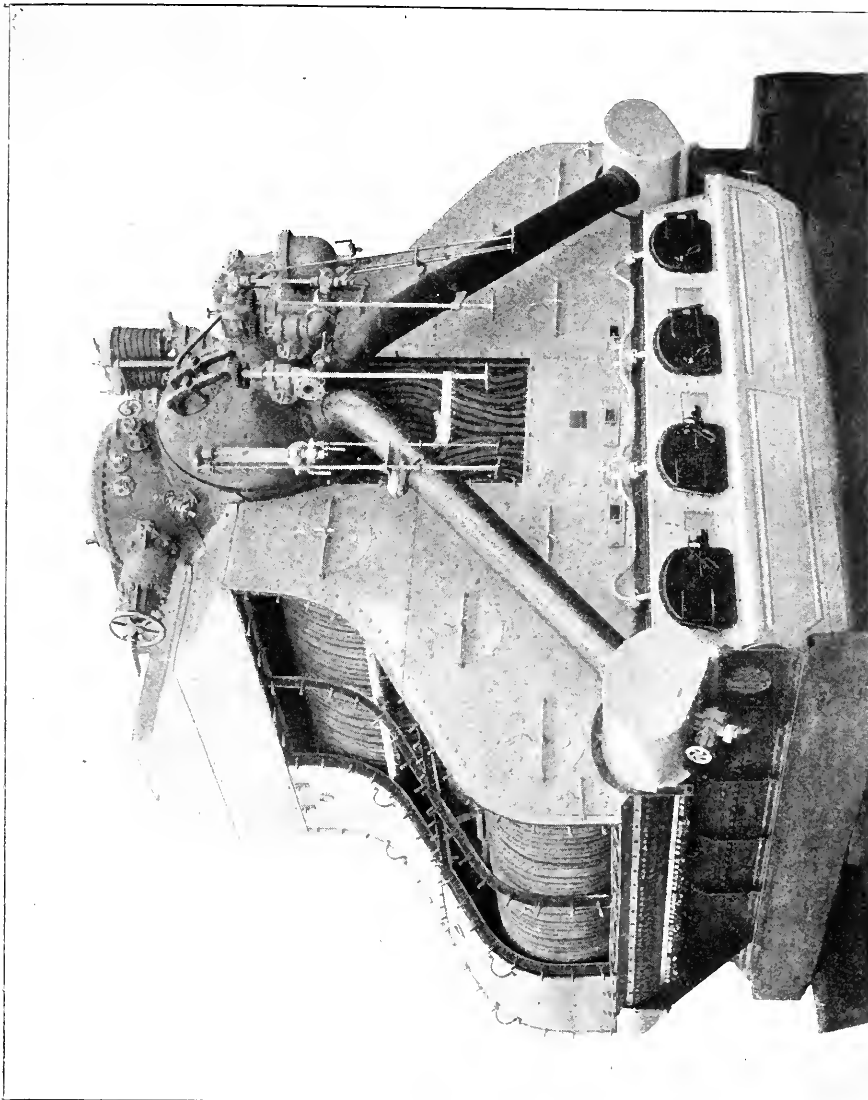


Fig. 231.—Reed's Patent Water-tube Boiler for Cruisers, Scouts, and Destroyers.

Mr. J. W. Reed, of Jarrow. It will be seen to consist of the usual steam receiver at top and the two wing water receivers at the bottom, both of which are in all essentials the same as found in the Normand, Thornycroft,

TABLE LVII.—Express Boilers. Particulars of Surface, Weight, &c.

Name of Ship.	Type of Boiler.	Inventor.	Number of Boilers.	Total Grate Area.	Total Heating Surface.	Ratio Heating Surface to Grate.	Working Pressure.	Total Weight of Boilers, &c.	Total Weight per Sq. Ft. of Grate.	Total Weight per I.H.P.	Total I.H.P. on Trial.	Air Pressure in Stokehold.	I.H.P. per Sq. Ft. of Grate.	I.H.P. per Ton of Boilers.	Coal per I.H.P. per Hour.	Coal per Sq. Ft. of Grate per Hour.	Heating Surface per I.H.P.
				Sq. Ft.	Sq. Ft.		Lbs.	Tons.	Lbs.	Lbs.		Ins.			Lbs.	Lbs.	Sq.Ft.
H.M. Ships:—																	
"Rattlesnake,"	Loco.	Laird.	2	126	4,639	36·8	140	85·2	1515	70·0	2740	2·30	21·7	32·1	...	...	1·69
"Karakatta,"	,,	Belliss.	2	190	5,330	28·0	150	95·6	1127	55·7	3840	1·70	20·2	40·4	...	...	1·3
"Gossamer," -	,,	Sheerness.	2	183	5,654	30·8	150	103·4	1267	63·4	3654	2·70	19·9	35·4	...	...	1·5
"Speedy," -	Water tube.	Thornycroft.	8	204	17,700	86·3	...	107·2	1177	51·0	4703	1·70	23	43·9	...	...	3·7
"Havock," -	Loco.	Yarrow.	2	100	4,656	46·5	180	58·0	1299	37·0	3497	3·66	34·9	60·0	2·59	90·6	1·33
"Hornet," -	Water tube.	Yarrow.	8	154·4	8,216	53·2	180	49·0	711	28·0	3884	1·60	25·1	79·1	2·4	60·0	2·11
"Daring," -	,,	Thornycroft.	3	189	8,892	41·7	210	56·0	644	30·0	4208	3·37	22·2	75·0	3·49	77·8	2·11
"Ferret," -	,,	Laird. / Normand.	4	154	8,812	52·7	175	64·2	900	32·0	4484	4·10	29·9	69·7	3·88	113	1·96
"Janus," -	,,	Reed.	4	209	10,164	48·6	210	70·6	757	41·0	3885	2·39	18·5	54·9	3·15	58·6	2·61
"Haughty," -	,,	Yarrow.	8	179	8,203	45·8	200	62·1	777	33·0	4208	2·30	23·5	67·7	2·72	63·9	1·94
"Conflict," -	,,	White.	3	212·5	11,250	52·9	185	83·0	875	38·0	4931	4·20	23·2	59·4	...	...	2·32
"Sturgeon," -	,,	Blechynden.	4	176	10,022	57·0	200	62·2	791	32·0	4367	3·1	24·8	70·1	3·29	81·6	2·29
"Quail," -	,,	Normand.	4	210	11,945	56·8	220	75·0	800	26·0	6347	4·25	30·2	84·6	2·48	74·9	1·88
"Desperate," -	,,	Thornycroft.	3	196	11,855	60·5	220	66·0	754	25·0	5795	3·45	29·5	87·8	2·37	70·1	2·04
"Star," -	,,	Reed.	4	255	13,384	52·5	250	89·4	785	32·0	6318	3·32	24·7	70·6	2·53	62·7	2·11
"Gipsey," -	,,	Yarrow.	4	245	13,561	55·3	250	68	622	23·0	6523	4·4	26·6	95·9	2·51	66·8	2·07
"Avon," -	,,	Thornycroft.	4	236	13,400	56·8	230	71	674	26·0	6153	3·6	26·0	86·6	2·59	67·5	2·17

or Yarrow boilers, but in this case they are connected differently—viz., by tubes zig-zagged somewhat like those in the Du Temple boiler, but not to the same extent, nor is there any reduction in the diameter of these small tubes as in the Du Temple. Mr. Reed has, besides, methods of his own for securing the small tube ends in the tube holes, whereby a tube can be easily fitted and withdrawn; and, when in place, it is capable of a certain amount of self-adjustment without any tendency to leak. This is effected by screwing the ends of the tube and fitting them with nuts formed to suit the chamfering or spherical recessing at the mouth of the tube holes.

Of the "Express" boilers none have done better than Mr. Reed's, either in respect to the quantity of steam generated per square foot of heating surface or to the number of pounds of water evaporated per pound of fuel When severely forced, it seems to suffer little or no harm from the process, and at cruising speeds it is very economical. It, however, has the objection common to the Normand, Thornycroft, and certain other boilers, of having the tubes so considerably bent as to prevent examination internally.

Belleville Boiler (fig. 232).—This boiler had been for very many years a favourite in France, and fitted into many warships, as also into a large number of their merchant ships. It had also been used extensively in America, both for marine and land purposes, when it was at last more favourably looked upon in this country, the lead in the matter being taken by the Admiralty, who replaced the wet-bottom locomotive boilers of H.M.S. "Sharpshooter" with boilers of this kind; the first-class cruisers "Powerful" and "Terrible," each of 25,000 I.H.P., and many other ships, cruisers, battleships, and gunboats were afterwards fitted with them. For large ships this boiler seemed suitable, and can compete successfully with the other forms of water-tube boiler in the question of weight; for smaller sizes, however, several of the before-mentioned boilers have advantages over the Belleville. It was for some time the cause of considerable trouble and anxiety, and although, now when they are better understood and more carefully worked, they do good service but they are no longer fitted to new ships. It consists of the usual top drum and bottom feed collector, connected by what is virtually a set of flattened spirals; in other words, each element is of the form that a spiral spring would be if heated and flattened. There are, of course, the usual down-cast pipes for circulating, besides special apparatus for preventing priming, for feeding, and for reducing the pressure. This boiler, like most of its congeners, must, to be successful, be worked with steam at a high pressure, or, to speak more correctly, with steam of low volume per lb.; consequently, the manufacturers of it usually design it for a much higher working pressure than is required, and fit it with an ingenious, and fairly reliable, reducing valve. Each element is made up of a series of tubes of considerable diameter, extending the whole length of the boiler, connected at the ends by junction boxes of malleable cast iron (fig. 232*a*), to which they are screwed, and they are zig-zagged so as to form the flattened spiral. These front junction boxes have, opposite each tube end, a hole of sufficient size to admit of the thorough examination and cleaning of the tubes, and closed with an oval door clamped in the same way as a manhole door. The top of each element enters the upper drum with a short internal pipe, and inside the drum is a series of dash plates, so arranged as to thoroughly separate the water and the steam. There is further, however, external to the boiler, a separator, having an internal

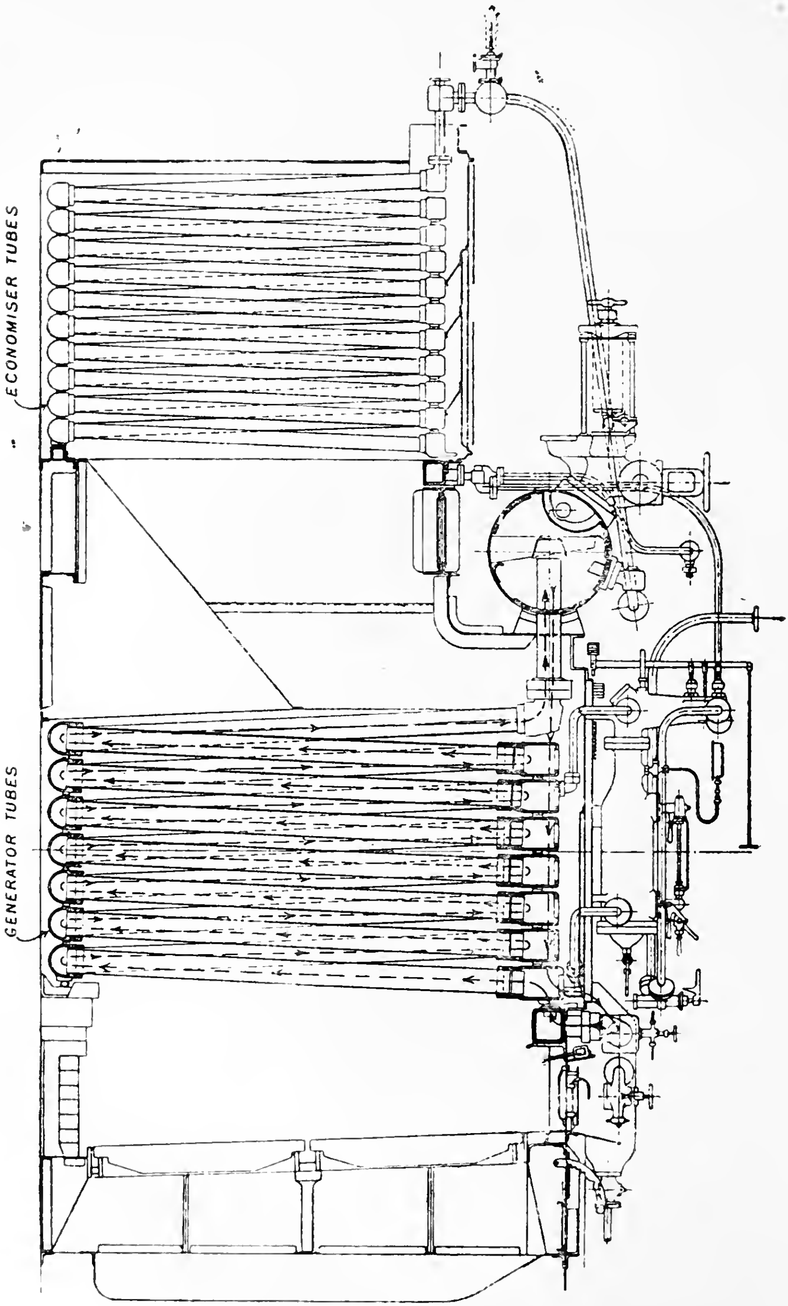


Fig. 232.—Modern Belleville Boiler, with Economiser

spiral dash plate, and, at the bottom, an automatic drain. This boiler is a most costly one to manufacture, and even when made with an accuracy exceeding that usually followed in engine building it loses so much water by the very large number of small leaks as to seriously reduce its efficiency, as well as to require a large reserve of feed water and a constant use of the distillers. After a most lengthy and carefully made series of experiments with it and the cylindrical boiler, the Committee appointed by Parliament condemned its use in H.M. Navy on these and other grounds; at the same time, it must be said that, while recommending that all large ships should have a few cylindrical boilers for cruising and use in port, certain other forms of water-tube boilers should be further tried and worked tentatively with the object of using them in such ships solely when high speed is necessary. Now water-tube boilers are exclusively used.

The boilers so recommended, besides the Babcock & Wilcox, and the Yarrow with large tubes already described, were the Dürr boiler and Niclausse boiler, both well known and used on the Continent; the latter in the French

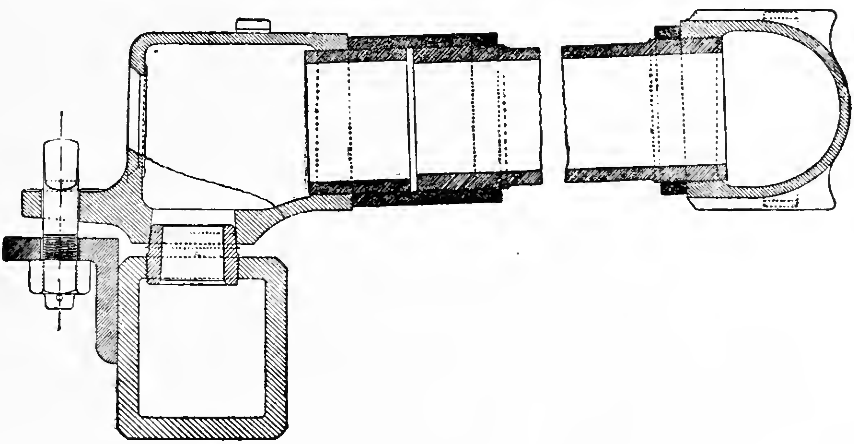


Fig. 232*a*.—Tube and Headers of a Belleville Boiler.

and the former in the German Navy are favourites. Each of them has its origin in the patents of Jacob Perkins of 1831, and Howard of 1869. Perkins adopted an internal tube to cause a circulation in the large one exposed to fire; this idea was enlarged and improved by Field, by whose name such tubes are known instead of by Perkins. Howard placed his large tubes at a small angle to the horizontal and connected their front ends to a vertical chamber, and fitted them with concentric circulating tubes as far as the water level. In 1870 Miller improved on this by fitting a vertical division plate in the chamber, into which plate he fitted the circulating tubes, and placed on top of the chamber a horizontal cylindrical receiver, with which both divisions of the vertical chamber had easy communication, so that the water flowed down into the front or outer part of it and through the circulating tubes into the main tubes, and from them into the back or inner part of the chamber and thence to the receiver, where it gave up the steam, and flowed back again over the same course.

TABLE LVIII.—H.M.S. "SHARPSHOOTER." SPECIAL SEA TRIALS. BELLEVILLE BOILERS.

No. of Trial.	Date.	Distance Run.	Duration of Trial.	Nature of Trial.	Indicated Horse-Power.	Speed by Log.	Number of Boilers in Use. (Total Number, 8.)	Grate Surface in Use.	Total Heating Surface in Use.	Coal. Total during Trial.	Coal. Per Sq. Ft. of Grate per Hour.	Coal. Per I. H. P. per Hour.
	1895.	Miles.	Hours.					Sq. Ft.	Sq. Ft.	Tons.	Lbs.	Lbs.
1	June 20 to 22,	693	$48\frac{1}{2}$	Sea	1553	14·3	6	202	5770	63	14·4	1·87
2	June 25 to 28,	1000	69	Sea	1549	14·4	6	202	5770	91·4	14·6	1·91
3	July 3 to 6,	1000	$66\frac{1}{2}$	Sea	1569	15·0	6	202	5770	96·5	16·0	2·07
4	July 11 to 14,	1000	68	Sea	1554	14·7	6	202	5770	91·8	14·9	1·94
...	July 19 to 23,	Trial abandoned on account of bad weather.										
5	July 28 to 31,	1000	$66\frac{3}{4}$	Sea	1804	14·9	6	202	5770	103·4	17·2	1·92
6	August 7 to 11,	1000	$69\frac{1}{2}$	Sea	1820	14·4	6	202	5770	104·2	16·6	1·84
7	Sept. 18 to 21,	1000	65	Sea	2016	15·3	6	202	5770	105·1	17·9	1·8
8	Sept. 28 to October 1,	1000	$63\frac{1}{2}$	Sea	2037	15·7	6	202	5770	103·2	18·0	1·79
9	October 8 to 13,	1000	62	Sea	2194	16·1	6	202	5770	105·4	18·8	1·73

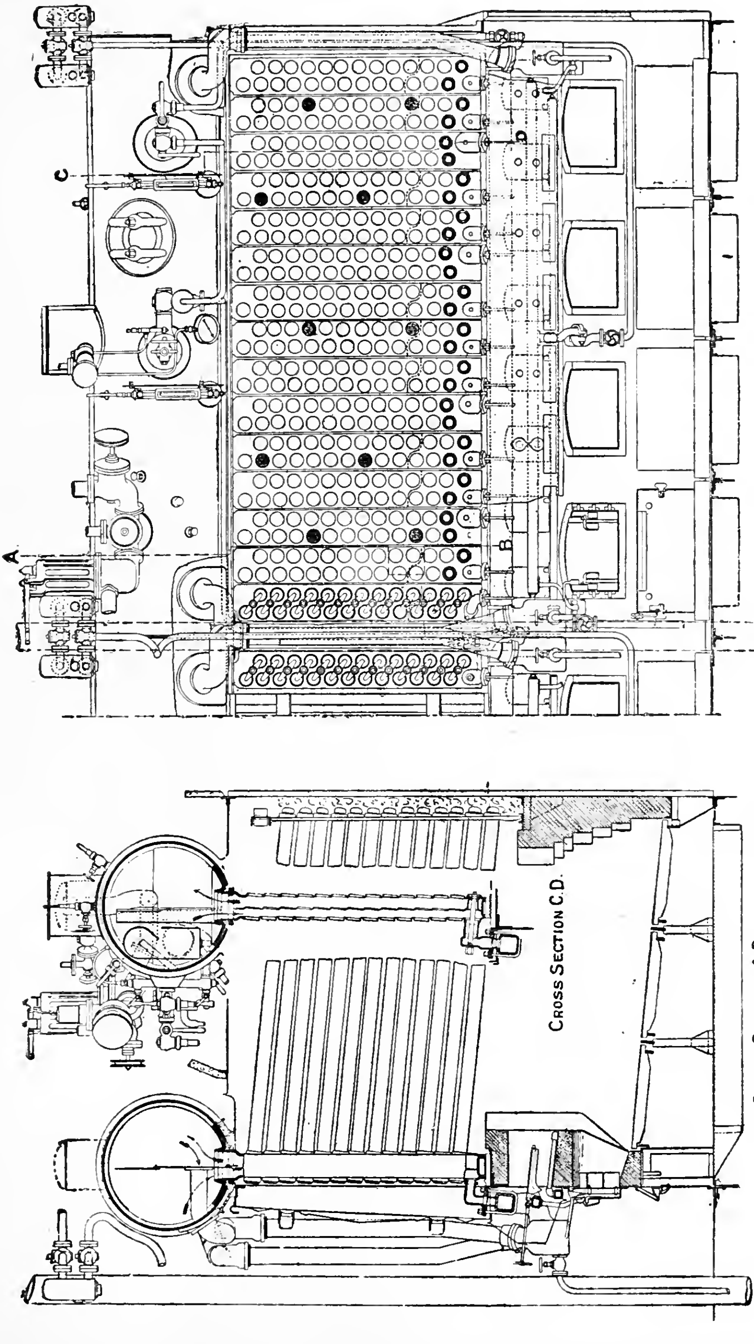


Fig. 233.—Niclausse Modern Marine Boiler.

The Dürr Boiler is substantially the above, with improvements in detail.

The Niclausse Boiler (fig. 233) differs from both Howard's, Miller's, and Dürr's in having a series of vertical boxes or headers fitted with a division plate, and having properly formed mouthpieces to allow of the tubes having a considerable slant while they themselves are vertical. These headers are made of steel or malleable cast iron, and connected to the under side of the steam receiver by neck pieces. The tubes in this case pass through from front to back of the headers, and fit quite steam-tight in a slightly-taper hole in the division plate and back. The portion of the tube between these is pierced with large oval holes to permit the water passing freely from it to the inner division of the header. The inner or circulating tube is fitted through a hole in the plug fitted in the end of the large tube (fig. 233*a*), and held in place by a neck piece which fits through and steam-tight in a hole in the stopper of the hole of the larger tube. The back ends of the big tubes are reduced in diameter, screwed, and fitted with cap nuts, which are taken

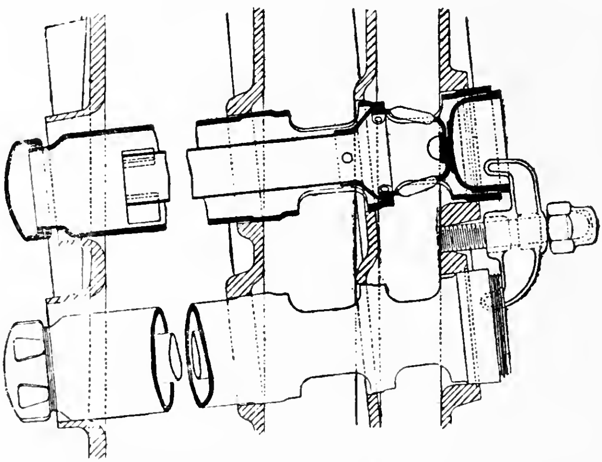


Fig. 233*a*.—Niclausse Tubes as fitted in place.

off when the boiler is to be drained dry; therein lies one of the chief objections to this and the Dürr boiler, as the time taken to remove and replace so many cap nuts is serious in all cases, and especially so in ordinary mercantile ships with a limited staff. So long as distilled water is used to make up waste there will be little need to remove them for cleaning, and it would only be in naval ships where so much harbour service would require frequent drying out.

The Niclausse boiler, as shown in fig. 233, is a favourite one in France, and used in the French Navy; it has also been used extensively in the Navies of the United States of America and of Japan. It possesses several noticeable features, but perhaps the strongest one to recommend its use is that there is a positive and constant circulation of water in each and every tube, and a steady transmission of the steam generated in them to the steam chest without in any way interfering with it. In the English Navy it has proved to be an economic and reliable generator of steam, and in that respect compares favourably with the other types of water-tube boilers, and when the

TABLE LIX.—RESULTS OF STEAM TRIALS WITH NICLAUSSE BOILERS.

H.M. Ships.	"Berwick."	"Suffolk."	"Fantome."	"Clio."	"Carnarvon."	"New Zealand."
FULL POWER TRIALS—						
No. of Niclausse Boilers,	34	34	4	4	22	18
Total heating surface, sq. ft.	52,314·31	52,314·31	4,000	4,000	43,566·20	45,640·47
Total grate surface, "	1,657·39	1,657·69	134·98	134·98	1,224·00	1,416·03
Ratio H—G,	31·5	31·5	29·63	29·63	35·59	32·23
Boiler pressure, lbs. per sq. in.	298·83	298·83	..	..	210·0348	..
External diameter of tubes, inch	$3\frac{5}{16} \times 2\frac{31}{32}$	$3\frac{5}{16} \times 2\frac{31}{32}$	$3\frac{7}{32} \times 2\frac{31}{32}$	$3\frac{7}{32} \times 2\frac{31}{32}$	$3\frac{5}{16} \times 2\frac{31}{32}$	..
Contract I.H.P.,	22,000	22,000	1,400	1,400	21,000	18,000
Contract speed, knots	23	23	13·25	13·25	22·25	18·5
Engine power at full power run, I.H.P.	22,680	22,645	1,453	1,435	21,489	18,383
Duration of trial, hours	5	8	8	8	30	8
Speed obtained, knots	23·613	24·7	13·5	13·3	23·3	18·59
Coal consumption per sq. ft. of grate hour lbs.,	26·02	30·02	16·49	21·61	..	..
" " I.H.P. " "	1·91	2·19	1·52	2·03	2·29	2·06
REDUCED POWER TRIALS—						
Engine power at duration trial, I.H.P.	16,622	16·350	1,020	1,041	15,212	12,918
Duration of trial, hours	30	30	30	30	30	30
Speed, knots	21·644	21·2	12·5	13	21·43	16·9
Coal consumption per sq. ft. of grate hour lbs.,	17·93	18·64	11·55	14·75	..	..
" " I.H.P. " "	1·78	1·89	1·52	1·94	1·96	1·84
Engine power at slow run, I.H.P.	4,671	4,954	339	311	4,756	3,958
Duration of trial, hours	30	30	30	30	30	30
Speed, knots	14·59	14·4	9·25	8·7	14·3	9
Coal consumption per sq. ft. of grate hour lbs.,	6·20	6·59	5·53	..	..	..
" " I.H.P. " "	1·75	1·89	1·71	2·32	1·78	2·02

fleet ran from Gibraltar to Quebec as a trial of endurance, etc., the following facts are worth recording :—

			Tons.
Coal consumed by H.M.S.	" Bedford "	with Belleville boilers was	1,378
,, ,,	" Essex "	,, ,, ,,	1,334
,, ,,	" Cornwall "	,, Babcock & Wilcox boilers was	1,245
,, ,,	" Cumberland "	,, Belleville boilers was	1,223
,, ,,	" Berwick "	,, Niclausse ,,	1,161

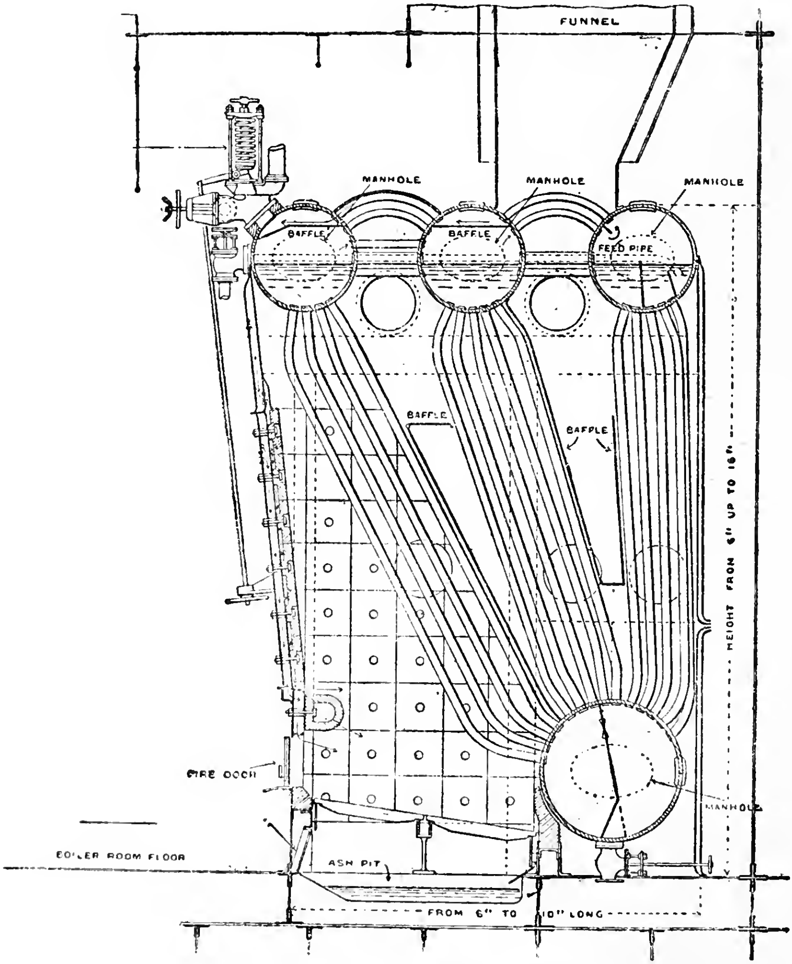


Fig. 234.—Stirling Boiler (Marine Design).

Table lix. gives some results of trials with these boilers in various ships of the British Navy, which are altogether quite satisfactory considering that the evaporation is equal to about 10 lbs. of water per lb. of coal at full power. The water consumption of county class of cruiser per I.H.P. is very high at full power.

The Stirling Boiler (fig. 234), which bears a strong family likeness to Rowan's boiler without its defects, is now coming into use for marine work after having given satisfaction to its users on shore. It has the merit of simplicity, and practically straight tubes of good size placed nearly vertically.

The Hohenstein Boiler.—This boiler (fig. 235) is a favourite one in the

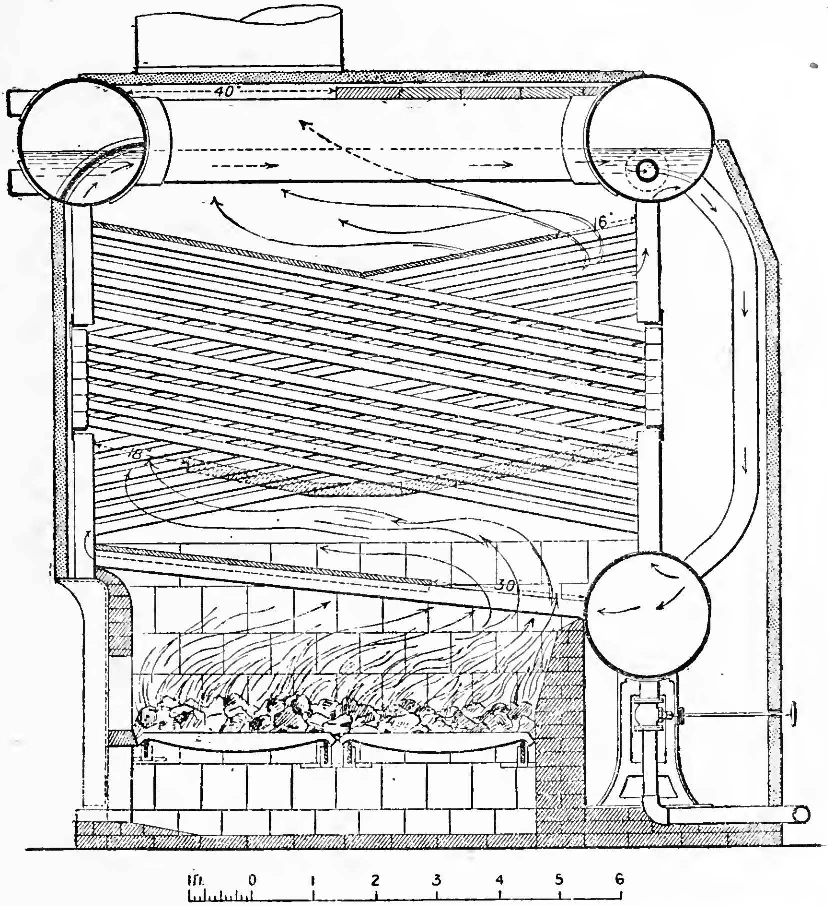


Fig. 235.—Hohenstein Boiler.

United States, for general use in the Navy of that country. It has been experimented upon very considerably by United States officials, and with good results. It consists essentially of two steam receivers at top, parallel with one another, and two water receivers at bottom, immediately below the steam receivers and connected to them by downcast pipes; or there may be only one at the back, as shown in fig. 235. Extending above these water receivers are rectangular section chambers or headers, and similar chambers extend from and below the steam receivers. These chambers are connected by a pair of pipes placed diagonally across the fire and connected at their remote ends by a junction box. The water, therefore, circulates by rising from the water receivers into the chambers, passing upwards through the tubes connected to it through the junction boxes, and so through the second set of tubes to the chambers underneath the steam receivers, and from them into the steam receivers themselves, and. finally, back again by the downcasts to the water receivers. It will be seen that the circulation is positive and regular. The tubes thoroughly intercept the heat in its passage from the grate to the chimney. They are straight, and easily examined, cleaned, or removed.

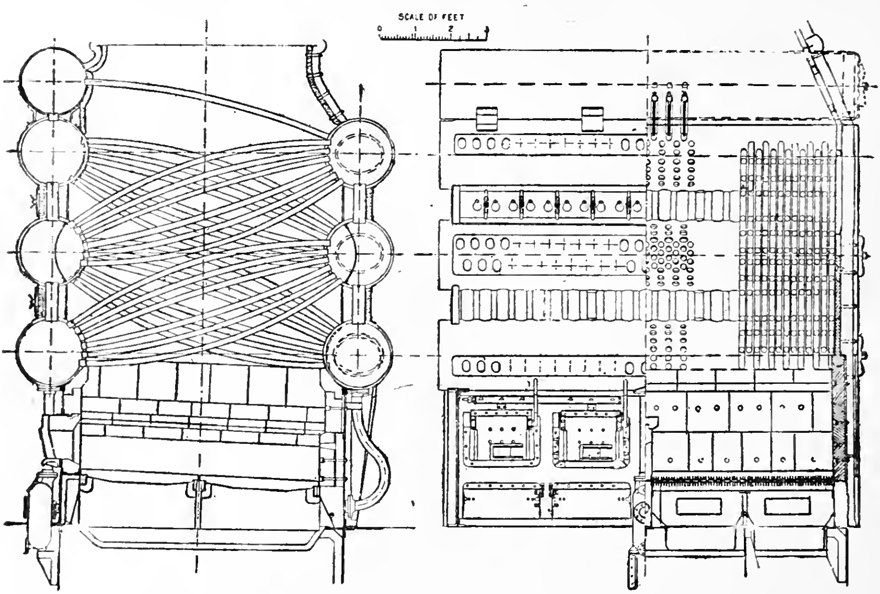


Fig. 236.—Miyabara Boiler of I. Japanese Navy.

The Miyabara Boiler (fig. 236), the invention of Admiral Miyabara of the I. Japanese Navy, is ingenious and somewhat like the Hohenstein, but consists of two sets of cylindrical drums placed parallel, and some 6 to 8 feet apart; each set consists of 3, one above the other. The middle drum of each is connected to the top and bottom drums of the opposite set by tubes 2 inches to 3 inches diameter. The top drum in each set is connected to the bottom one by downtakes. It is very simple, and easy to make, having nearly straight tubes with a good inclination; the hot gases will be well broken up and distributed, and limited spaces are provided for them to come in contact with the outer casing. It is now used in the cruisers and battleships of Japan, and the following is given as the results of experiments:—

TABLE LX.—Trials of Miyabara Boiler.

	No. 1.	No. 2.	No. 3.	No. 4.
Type of boiler, ...	Miyabara	Miyabara	Miyabara	Miyabara
Nature of trial, ...	Coal only	Coal only	Coal only	Coal only
Duration of trial, . . in hours	3	3	8	3
Grate area (G.S.), . . in sq. ft.	53·906	53·906	53·906	53·906
Heating Surface (H.S.), . ,,	1911·77	1911·77	1911·77	1911·77
Ratio of $\frac{H.S.}{G.S.}$, ...	35·466	35·466	35·466	35·466
Barometer, . . . in inches	29·990	29·818	30·123	29·990
Boiler pressure, . . in lbs.	233·84	233·95	235·88	233·00
Total fuel burnt per square foot of grate per hour, . in lbs.	17·721	29·535	37·411	44·302
Total fuel burnt per square foot of heating surface per hour, in lbs.	0·4997	0·8328	1·0548	1·2492
Quantity of ash and clinker during trial, . . . in lbs.	210·729	373·959	767·966	285·714
Kind of fuel, ...	Welsh coal	Welsh coal	Welsh coal	Welsh coal
Temperature of feed water, at F. °	75·05	75·00	73·33	74·63
Temperature of Atmosphere: Outside of boiler room, at F. °	79·68	85·37	83·16	80·95
Temperature of Atmosphere: Inside of boiler room, . at F. °	91·47	91·16	89·84	92·37
Water evaporated from and at 212° F. per lb. of fuel, . in lbs.	10·439	9·2646	9·1603	8·2293
Water evaporated from and at 212° F. per square foot of heating surface, . . in lbs.	5·2162	7·7150	9·6627	10·2800

The Thornycroft-Marshall Boiler (fig. 237) is also one with large tubes, that seems a good design for marine or land. Its sectional form, which is a good one for ship use, consists of a steam drum, with headers standing vertically downwards, capable of taking two vertical rows of tubes, and spaced so that similar headers standing upwards from the feed collectors or water drums may fit between them. The bottom hole in the lower header is fitted with a tube, whose back end fits into a junction box like that of a Belleville boiler; another tube is fitted into this box, and the bottom hole of the header from the steam drum, and so makes the connection from the feed to the steam receiver; and so on with the other elements, as shown in fig. 237, like those in the Hohenstein. Here, too, the tubes are straight, and at a good angle; they can be readily examined and replaced or cleaned, and the circulation is certain and good. This boiler can also be readily drained and cleaned.

There are some other boilers worthy of notice in a special treatise, but not, however, of the same interest to the marine engineer as the foregoing.

The water in the water-tube boiler is, as a rule, very considerably less than that in the ordinary boiler, and to this fact is in no small measure due the exceedingly light weight of certain boilers of this type. To the designer

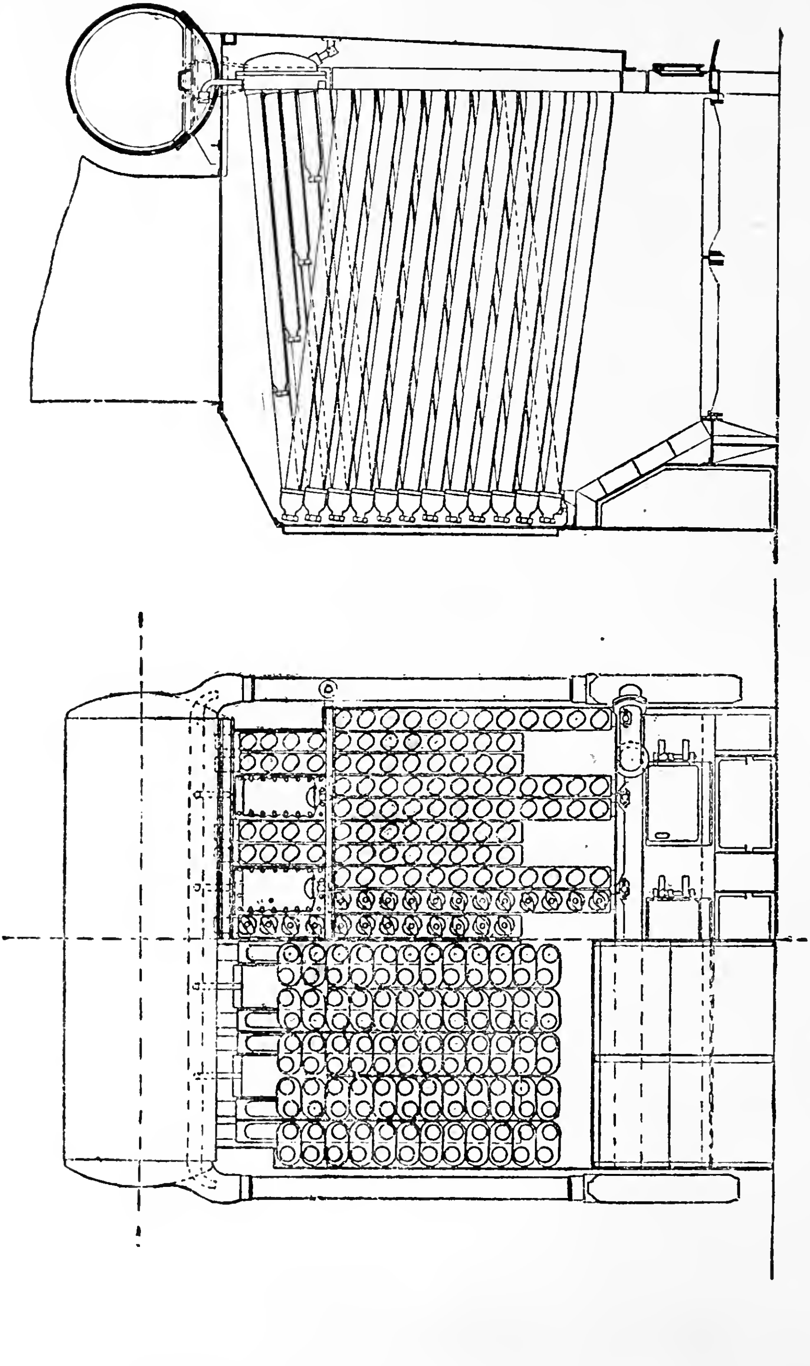


Fig. 287.—Thornycroft & Marshall's Boiler (Large Tube type).

this is of course a great advantage, and in case of accident, either from within or from without—from a shot, for example—the possible amount of harm is limited, especially if there be no obstruction with the draught; but to the firemen it is otherwise, being a source of trouble, as it permits of such great variations in steam pressure from temporary causes. The automatic reducing valve, as fitted to the Belleville and other boilers, tends to prevent any ill effect from this being experienced in the engine-room, for, so long as the pressure in the boiler does not fall below that at which the apparatus is set to deliver, there will be little or no variation at the high-pressure valve-box.

The total weight in the boiler-room includes, besides the boiler proper, the water, mountings, and fittings; the furnace fittings, brickwork, etc.; the casings and uptakes to the funnel base; in fact, everything is included that is necessary to make a fair comparison. Taking naval boilers of the old double-ended type, the average weight of water is about 29 per cent. of the total weight; while with single-ended naval boilers, it is about 26 per cent. The steam pressure for which these boilers were designed is, however, only 155 lbs. With double-ended boilers made in accordance with the Board of Trade Rules for pressures of 160 lbs., the water was about 33½ per cent. ot the total weight, and about 32 per cent. with the single-ended boilers. With the gunboat type of boiler, the water is 33 per cent. when designed in accordance with the Admiralty Rules. With the Belleville boilers the water was only about 8 per cent. of the total weight; with the Babcock & Wilcox boilers it is about 14½ per cent.; with the Yarrow boiler it is about 15½ per cent.; with the Thornycroft boiler 15 per cent.; and with the Normand boiler about 24 per cent. The dry-bottom locomotive boiler, which was the rival of the smaller class of water-tube boiler, has water to the extent of 30 per cent.

The total weight per I.H.P. at forced draught was on the average with the ordinary old naval double-ended boilers about 88 lbs., and with the single-ended ones 108 lbs. The lightest of all was 80·6 lbs. The Belleville and the Babcock & Wilcox are both under 80 lbs., and with an air pressure of 1 inch of water the weight is as low as 70 lbs. with the Belleville. Taking the naval cylindrical boilers with natural draught the average weight per I.H.P. was 121 lbs. with double-ended boilers, and 133 lbs. with single-ended. It is true that the latter figure was for the larger sizes for battleships and cruisers. With the Belleville boilers the weight per I.H.P. is 82 lbs., and the Babcock & Wilcox 88·7 lbs.; and with the large allowance provided in the cruisers it was 110 lbs., or 11 lbs. less than the old double-ended boilers for 155 lbs. pressure, and 23 lbs. less than the single-ended boilers for 155 lbs. pressure as supplied to the Navy. It is manifest by these figures that there is a distinct saving in weight with the water-tube boilers in large ships.

It is, however, in the smaller class of vessels that the great advantage in using these boilers is experienced. Here types of boilers can be employed that would hardly be admissible in the larger ships * and the full advantage of the system can be utilised. By again referring to the table, it will be seen that with the locomotive boiler the weight per I.H.P. is 33 lbs., while with the Yarrow boiler the weight is only 25 lbs., considerably under the locomotive, as are all the other forms now fitted into torpedo boat destroyers. The great saving in each case is, of course, due in large measure to the small quantity of the water. Referring to Table lxv. it will be seen that, when taking all

* The Yarrow boiler, with somewhat larger tubes than formerly obtained, is now fitted in all classes of war ship and in large and small express passenger ships.

the weights which come into and are included in the boiler-room weights according to Admiralty Rule, for each ton with single-ended boiler, etc., 17·53 I.H.P. was developed by the engines; with double-ended boilers, 20·1 to 23·8; with Belleville boilers, 21·7 to 22·5; with Thornycroft, etc., in 3rd class cruisers, 43·1; with locomotive in torpedo gunboats, 29·5; with Babcock & Wilcox, in same ships, 32·5; with Niclausse, 26·3. With the express boilers in torpedo boat destroyers as much as 95·9 I.H.P. has been developed with Thornycroft's, 84·6 with Normand's, 79·1 with Yarrow's, and 70·6 with Reed's, as against 60 with locomotives in the "Havock."*

When the cylindrical boiler is designed in accordance with the Board of Trade Rules, the difference is more marked, for here the double-ended boiler is such that the total weight per I.H.P. is 149 lbs. when the working pressure is 160 lbs., and as much as 196 lbs. when it is 210 lbs. pressure. Single-ended boilers are, of course, heavier still. The Belleville boiler is only 107 lbs. per I.H.P. with a working pressure of 250 lbs., and the Babcock & Wilcox 115 lbs. for a pressure of 200 lbs.

The space occupied by these boilers does not differ materially from that required for cylindrical or locomotive boilers, and the stokeholes, as a rule, must be as long; but with some water-tube boilers a comparatively short stokehole is sufficient. The water-tube boiler is capable of subdivision to an extent not possible with the cylindrical, and in this respect it is in the hands of the designer much as the old box boiler was.

Feed Arrangements.—It is, of course, essential to the good working of these boilers, as of all boilers containing such comparatively small quantities of water, that the supply of feed-water shall be regular and delivered in such a way as to improve rather than to retard the circulation. To satisfy the first condition, it is necessary to have good feed-pumps and a good store of water, from which they may draw whenever required. The ordinary engine pump drawing from the hot-well is always more or less intermittent in its action. and while no great inconvenience is felt when it is supplying the ordinary tank boiler or the larger kinds of water-tube boiler, it would not suit the more delicate ones. It may, therefore, be taken, as a rule, that this class of boiler should have an automatic apparatus that effectively controls the feed. The Belleville boiler was always fitted with an arrangement which consists essentially of a chamber in which there is no disturbance by circulation or other cause, but so connected to the boiler that the water level in it is the normal level of the water in the boiler when "standing." It contains a "float" arrangement which operates in an ingenious, if somewhat complicated way, a controlling valve on the feed-pipe, so that, as the water falls in the boiler, the feed-water is allowed to pass, and as the water rises above the working level it is gradually shut off. Sir J. Thornycroft, Mr. Mumford, and others have somewhat similar methods for regulating the supply. Other engineers, following Mr. Yarrow, have, however, preferred to fit to each boiler an independent donkey pump, and to set it working with steam from its own boiler at such a speed as to just keep up the supply. This method, while convenient, cannot be said to be automatic; it might, however, be so if to the feed-valve, or even to the steam-valve, of the donkey is fitted a means of controlling by a similar apparatus to the above. As the variation of level in most water-tube boilers is somewhat limited, the "float" apparatus cannot have a great range, as may be seen in fig. 279.

* To-day with turbines and more modern designs of these boilers 25 per cent. better output of S.H.P. is obtained.

TABLE LXI.—H.M.S. "HERMES." WATER CONSUMPTION TRIALS.

BELLEVILLE BOILERS. 300 LBS. WORKING PRESSURE. MADE IN MAY AND JUNE, 1899.

Description of Trial.	Percentage of Full Power.	Steam Pressure.			Lbs. of Water per I.H.P. per Hour.			Lbs. of Coal per I.H.P. per Hour.	Remarks.
		Boilers.	Engines, Starboard.	Engines, Port.	Main Engines.	Auxiliary.	All Purposes.		
12 hours at 1,000 I.H.P.	10·0	241	169	176	21·9	5·2	27·1	2·5	Main links linked up as far as practicable. Cut-off in H.P. cylinder 13·5 per cent.
12 ,, 1,000 ,,	10·0	163	124	123	20·1	6·67	26·8	2·3	Main links in full gear—independent links run in. Cut-off in H.P. cylinders 47·8 per cent.
30 ,, 2,000 ,,	20·0	196	126	127	17·06	4·22	21·28	1·76	
30 ,, 7,500 ,,	75·0	263	223	222	15·3	2·1	17·4	1·57	
12 ,, 7,500 ,, (*Closed exhaust system in use.*)	75·0	264	228	219	...	...	16·84	1·5	Auxiliary exhaust at 26 lbs. pressure in use on evaporators and surplus to L.P. receivers.
8 hours at 9,000 I.H.P. (*Closed exhaust system in use.*)	90·0	270	249	244	...	...	17·13	1·54	Auxiliary exhaust at 29 lbs. pressure in use on L.P. receivers only.
8 hours at 10,000 I.H.P.	100·0	240	231	228	15·64	2·05	17·69	1·58	

Report of Boiler Committee.

TABLE LXII.—H.M.S. "DIANA." WATER CONSUMPTION TRIALS, SHOWING COST OF AUXILIARIES.
CYLINDRICAL S.E. BOILERS. 155 LBS. WORKING PRESSURE.
MADE IN MAY AND JUNE, 1899.

Description of Trial.	Percentage of Full Power.	Steam Pressure.			Lbs. of Water per I.H.P. per Hour.			Lbs. of Coal per I.H.P. per Hour.	Remarks.
		Boilers.	Engines, Starboard.	Engines, Port.	Main Engines.	Auxiliary.	All Purposes.		
30 hours at 800 I.H.P.	8·3	101	96	97	20·59	5·68	26·27	2·82	
12 " 1600 "	16·6	124	120	121	18·94	4·21	23·15	2·52	Cut-off in H.P. cylinder 38 per cent.
30 " 1600 "	16·6	117	113	113	18·26	3·29	21·55	2·52	Cut-off in H.P. cylinder 44·5 per cent.
30 " 4800 "	50·0	133	128	128	17·09	2·14	19·23	2·02	
30 " 6400 "	66·6	138	129	129	17·9	2·56	20·46	2·38	
8 " 8000 "	83·3	150	144	144	17·75	1·94	19·69	2·39	

Report of Boiler Committee.

TABLE LXIII.—Results of Trials with Various Modern Marine Boilers in Lbs. per Hour.

Ship, Service, &c.	A North Atlantic Express.	B North Atlantic Pass.	C Eastern Pass.	D Eastern Cargo & Pass.	E Cargo & Pass.	F General Cargo.	G South Atlantic Passngr.	H Channel Express.	J Channel Express.	K U.S.A. Battle-ship.	L U.S.A. Cruiser.	M Scout.	N Scout.	P Scout.
Engines, Kind of,	Tur-bines	Recip. & Turb.	Quad. Recips.	Quad. Recips.	Triple Recip.	Triple Recip.	Triples & Turbs.	Tur-bines.	Grd. Turbs.	Tur-bines	Tur-bines	Tur-bines	Tur-bines	Turbs & Recips.
Horse-power developed,	68,850	13,950	11,530	7,000	8,598	1,790	10,840	6,670	6,100	24,100	20,904	17,800	17,200	17,250
Boilers, Kind of,	D.E. cylinder	D.E. cylinder	S. & D.E. cylinder	D.E. cylinder	D.E. cylinder	S.E. cylinder	S.E. cylinder	D.E. cylinder	D.E. cylinder	Yar-row	Bab-cock	Yar-row	Thorny-croft	White Foster
,, Total heating surface,	158,352	29,300	30,568	18,188	19,635	5,824	20,965	12,985	10,221	48,000	32,118	16,600	15,860	21,600
,, Working pressure, lbs.,	195	210	215	215	200	180	180	160	160	295	210	250	265	260
,, Draught, Nature of,	Howden Fd.	Funnel	Howden	Howden	Howden	Funnel	Howden	Forced	Forced	Funnel	Forced	Forced	Forced	Forced
Fuel, Description of,	W. coal	W. coal	W. coal	W. coal	Coal	Coal	Coal	W. coal	W. coal	Texas oil	U.S. coal	R.N. oil	R.N. oil	Oil
,, Calorific value, B.T.U.,	14,800	14,200	14,800	14,800	15,160	14,500	14,200	14,800	14,800	19,500	15,200	19,000	19,000	19,470
Ratio of T.H. surf. to grate area,	39·1	37·7	41·1	41·3	41·3	45·5	43·6	38·5	33·7	..	44·9	..	..	..
,, ,, horse-power,	2·31	2·10	2·65	2·60	2·27	3·80	1·94	1·95	1·68	1·99	1·54	0·93	0·92	1·25
Fuel consumed per sq. ft. of grate,	24·3	25·9	20·2	19·9	26·4	21·0	26·3	33·8	27·0	..	29·5	..	..	..
,, ,, T.H. surf.,	0·632	0·687	0·493	0·482	0·640	0·462	0·641	0·878	0·800	0·664	0·657	1·120	1·110	1·016
,, ,, horse-power,	1·43	1·44	1·300	1·25	1·45	1·70	1·24	1·71	1·34	1.32	1·60	1·033	1·025	1·27
(a) Water evaporated per sq. ft. T.H.S.,	5·55	6·67	5·24	5·19	7·05	3·98	6·27	8·77	8·54	7·73	8·72	12·84	13·24	13·49
,, ,, horse-power,	14·93	14·00	13·90	13·50	15·88	15·18	12·12	17·10	14·30	15·40	13·42	12·05	12·27	16·65
(b) ,, ,, lb. of fuel,	10·44	9·71	10·7	10·02	10·97	8·91	9·79	10·0	10·7	11·65	10·73	11·70	11·92	13·08
Comparative capability=A × B,	58·0	78·4	56·1	52·0	84·7	35·5	61·4	87·7	91·4	90·1	93·6	175·6	187·3	201·2
Efficiency of boiler per cent.,	71·7	78·5	73·3	70·0	76·5	62·4	80·0	70·0	70·0	63·0	71·3	71·6	71·7	74·9

TABLE LXV.—Comparison of Early Water Tube with Tank Boilers on Trials of Various Ships.

Class of Ship,	Battleships.		First-Class Cruisers.		Second-Class Cruisers.			Third-Class Cruisers.		Torpedo Gunboats.			
Name,	Mean of Six.	Mean of Six.	"Edgar" Class.	"Diadem" Class.	"Diana" Class.	"Apollo" Class.	"Arrogant" Class.	"Pearl" Class.	"Pelorus" Class.	"Alarm" Class.	"Speedy."	"Shel-drake."	"Seagull."
Indicated horse-power,	12,414	13,500	12,851	16,961	9,846	9,274	10,240	7,469	7,152	3,665	4,703	4,050	3,546
Boiler pressure,	149	289	146·2	286·5	151	155	265	155	253	155	...	151	145
Air pressure in stokehole,	0·85	0·19	0·7	0·305	1·06	0·9	0·06	1·4	2·9	2·20	1·70	0·5	0·16
Type of boiler,	Cylindrical.	Belleville.	Cylindrical.	Belleville.	Cylind. S.E.	Cylind. D.E.	Belleville, old.	Cylind. D.E.	Express.	Locomotive.	Thornycroft.	Babcock.	Niclausse.
Grate area, sq. ft.	817	1,050	812	1,460	624	575	869	410	353	171	204	252	276
Total heating surface, ,,	25,233	33,700	24,908	40,990	18,579	15,641	25,606	11,025	20,508	6,204	17,700	9,103	7,932
Total weight of boilers, &c., tons	724	623	646	767	548	434	463	314	172	124·4	132	124·8	1,347
Ratio heating surface to grate,	30·8	32·1	30·7	28·1	29·8	27·2	29·4	26·9	58·0	36·3	86·8	37·4	28·8
Coal per I.H.P. per hr., lbs.	2·29	1·72	1·70 (*a*)	1·85	2·18 (*b*)	...	2·00 (*c*)	...	2·36 (*d*)	...	...	1·57	2·11
,, sq. ft. grate per hour, ,,	34·8	...	...	...	...	...	...	...	..	...	...	25·6	27·1
Weight of boilers, &c., per I.H.P., ,,	131	103	111	101	124	104	99	98·6	52·9	76·0	62·9	69·9	85·1
Weight of boilers, &c., per sq. ft. of grate, ,,	1,971	1,329	1,687	1,172	1,959	1,674	1,170	1,715	1,092	1,630	1,450	1,109	1,093
I.H.P. per sq. ft. of grate,	15·2	12·9	15·2	11·6	15·79	16·1	11·82	18·2	20·9	21·4	23·0	16·0	12·8
,, ton of boiler,	17·1	21·7	20·1	22·1	17·97	20·7	22·5	23·8	43·1	29·5	39·6	32·5	26·3
Total heating surface per I.H.P.,	2·0	2·5	1·9	2·41	1·88	1·65	2·49	1·47	2·87	1·70	3·7	2·3	2·24
Total weight of machinery, tons	1,341	1,290	1,161	1,540	915	740	825	539	383	211	212	205	2·14
I.H.P. per ton of machinery,	9·3	10·5	11·1	11·0	10·76	12·5	12·34	13·8	18·92	17·3	22·1	19·3	16·9
Piston speed, ft.	875	918	923	923	937	910	906	880	981	875	875	875	...
Coal consumption per I.H.P. at sea trial,	2·0	2·36	2·0	2·33	2·11	...	2·36	...	...	...	...	2·14	2·15
I.H.P. on sea trial, 1 ship,	8,029	8,163	8,824	13,517	6,694	...	6,273	...	...	...	...	2,775	2,773

(*a*) At 10,517 I.H.P., (*b*) at 8307 I.H.P., (*c*) at 7275 H.P., (*d*) at 5388 H.P.

TABLE LXVI.—Early Trials of Various Boilers (Marine).

Type of Boiler.	Ratio Total Heating Surface to Grate.	Duration of Trial.	Kind of Fuel.	Fuel per Hour. Total	Fuel per Hour. Per Sq. Ft. of Grate.	Ash.	Temperature. Feed Water.	Steam Pressure.	Air. Pressure.	Air. Temperature.	Water Evaporated. Per Hour.	Water Evaporated. Per Lb. of Fuel.	Equivalent from and at 212°. Per Hour.	Equivalent from and at 212°. Per Lb. of Fuel.	Equivalent from and at 212°. Per Sq. Ft. of Heating Surface	A × B.	Remarks.
		Hrs.		Lbs.	Lbs.	%	° F.	Lbs.	Ins.					A.	B.		
Cylindrical, Single End.	22·3	3·75	Welsh.	418	19·8	2·81	61	70	...	...	3,190	7·64	3,800	9·09	7·60	69·1	Coal-testing trial.
Locomotive,	70·0	...	,,	440	35·5	5·70	...	130	...	...	...	...	5,370	12·20	6·20	75·6	By Donkin & Kennedy.
Belleville (original),	28·5	8·0	,,	2623	19·4	...	50	194	...	...	...	9·07	29,115	11·10	7·56	83·9	Contract quantity trial.
,, ,,	28·8	4·0	,,	1309	27·3	...	42	275	0·75	...	...	7·60	12,305	9·40	8·63	81·1	,, ,,
Niclausse,	34·5	7·27	Pwll Dffrn.	254	13·5	6·74	60	158	0·20	67·4	...	8·67	2,659	10·47	4·09	...	By Profs. Kennedy and Unwin.
,,	34·5	2·95	,,	482	25·6	6·74	55	144	0·56	62·3	...	8·53	4,984	10·34	7·67	79·3	,, ,,
Babcock & Wilcox,	37·4	3·0	,,	1260	20·0	...	115	200	0·10	...	11,483	9·11	13,784	10·94	5·85	...	Admiralty trial.
,, ,,	37·4	3·0	Nixon's Nav.	2564	40·7	...	70	185	0·25	...	19,577	7·63	23,460	9·15	10·00	91·5	,, ,,
Stirling,	48·5	5·0	Welsh.	524	16·0	6·98	102	198	...	...	5,433	10·30	6,314	12·05	3·95	...	
,,	48·5	7·5	,,	700	21·0	7·00	100	195	...	...	6,600	9·47	7,756	11·08	4·85	...	
,,	48·5	4·0	,,	994	3[illegible]·1	8·00	100	200	...	...	8,857	8·92	10,377	10·44	6·49	67·7	
Haythorn,	63·0	...	,,	102·6	15·1	...	47·2	196	Nat.	60·0	909	8·86	1,114	10·86	2·60	...	By Prof. Watkinson.
,,	63·0	...	,,	271·3	39·8	...	52·0	240	2·0	80·0	2,565	9·45	3,147	11·60	7·35	85·3	,, ,,
,,	63·0	...	,,	504	74·1	...	55·6	251	2·6	61·0	3,648	7·24	4,486	8·90	10·48	93·3	,, ,,
Express:— Reed's,	67·6	...	,,	1684	19·1	...	...	...	Nat.	...	...	...	7,969	11·60	2·30	...	
,,	33·8	...	,,	3344	46·6	...	...	...	1·5	...	...	...	31,551	9·42	9·20	86·7	
,,	33·8	...	,,	...	79·4	...	...	...	3·5	...	...	...	47,217	8·28	13·80	114·2	
Thornycroft,	49·0	3·0	,,	3024	67·2	...	56·0	225	3·0	50·0	21,295	7·04	25,976	8·59	12·20	107·8	Contract quantity trial.
Mumford's,	45·0	4·0	,,	...	25·0	...	56·0	175	0·15	...	...	9·00	...	10·90	...	...	Admiralty trial.
Seaton's,	25·8	6·0	,,	178	17·45	3·27	56·0	200	0·32	...	1,356	7·61	1,648	9·26	6·27	58·1	Coal test.
,,	25·8	3·0	,,	461	45·5	1·50	56·0	200	2·00	...	3,237	7·02	3,900	8·46	14·83	125·5	,,
Yarrow,	55·7	...	,,	627	19·0	...	...	180	...	...	...	...	7,524	12·00	3·00	...	Prof. Lambert and Mr. Haddon.
,,	55·7	...	,,	1716	52·0	...	...	180	...	...	...	...	17,675	10·30	9·50	97·9	,, ,,

Mr. Yarrow, however, adopted an ingenious arrangement for controlling the feed donkeys which is at the same time very simple and effective. There is an internal steam pipe in the steam receiver specially fitted for the feed donkey; its open end is surrounded by a small trough, and is at the highest working level of the water. (The trough is shaped like an inverted cone, and so keeps the water fairly smooth within it.) If the water rises above this point, water enters the steam pipe, gags the pump, and is carried through its exhaust pipe to the condenser or hot-well tank.

It is a disputed point whether to admit the feed-water into the upper or lower chamber; but if the principle of aiding and not obstructing circulation is observed, there need be little or no controversy on this point. Each case must be dealt with on its own merits; but, as a general rule, unless the feed-water is well heated, it is better admitted to the bottom chamber or at the downcast part of the upper chamber in a direction of the downward flowing currents; by preference the latter, as it is more likely to aid circulation. In this case, the feed-pipe may with advantage be made small, so that the water is injected at a high velocity. When the water is well heated before admission to the boiler proper it is not of much importance where it is taken, but an internal pipe with small outlets to insure distribution should be fitted. Mr. Yarrow caused the incoming feed to be heated by passing through the outer rows of tubes.

It is usual and advisable also to fit water-tube boilers with a settling chamber into which any solid matter admitted with the feed-water can be deposited and blown out. The principle on which these act is that solid matter of higher specific gravity than the water will deposit in an enlargement of a channel where there is a reduction in the velocity of flow; a change in direction of flow also helps to deposit. A certain amount of lime solution is used with advantage in these boilers, and a means for admitting it is necessary. As a rule, it may be introduced with the feed-water by placing it in the tanks.

In arranging casings and uptakes great care should be taken to insure a good distribution of the hot gases over the whole of the tubes. The tendency to shorten circuit is, of course, strong, and to prevent it considerable judgment is necessary in the placing of baffle plates and other devices. It is a matter of experiment, however, to so place them as to get the best results.

The fire-places in water-tube boilers are generally of such a nature as to require a brickwork erection; in fact, in the case of the Belleville boiler the brickwork was very extensive, but in the case of most of the others, and especially of the lighter kinds, the brickwork is limited to the immediate neighbourhood of the fire, the rest of the boiler up to the base of the funnel being enclosed with light ironwork, either lined with asbestos, tiles, or in some other way protected from the direct action of the heat of the gases. As far as possible it is better to use water walls than brickwork, both of which are only absolutely necessary to resist contact of the flame or direct radiation from the glowing fuel. Casing plates outside the tubes, not subject to the direct action of the fire, may be of very thin sheet steel, especially if stiffened with light angles or by corrugations; it is generally made portable, so that it can be easily removed for the cleaning of the outside of the tubes or the repairing of the boiler when necessary. Sheet asbestos $\frac{1}{4}$ inch thick forms a very good protection to these thin casings; it requires to be carefully secured and protected from damage, and for this purpose it is often completely covered by thin sheet steel, but this covering should be avoided if possible. Two layers of asbestos millboard with a space between them practically sealed, so as to prevent air circulation, makes a still better shield, and with care in fixing it will withstand a considerable amount of wear and tear. The Admiralty do not now require the fitting of dampers to the uptakes of water-tube boilers as found in the torpedo boat destroyers, and, indeed, the neoessity for them does not exist, while their presence may be a source of danger. The evaporation with these boilers is quickly checked by leaving the fire door open, or, much better still, by shutting off altogether access of air to the fire. Without the damper the bursting of a tube or other serious leakage is not dangerous, as the steam escapes immediately up the funnel. The Admiralty also require to be fitted an apparatus for drenching the fires with water, which can be operated both from the stokehole and on deck. The tubes do not get covered with soot and dust so badly as might be expected, and when Welsh coal is used in the boilers the deposit is easily removed. Oil fuel, however, is now almost exclusively used.

The life of these water-tube boilers is rather too short to be satisfactory, and the trouble experienced with the Belleville, due, however, in no small measure to the want of care and judgment by those in charge of them, caused them to be condemned. Others less in first cost, and not having such delicate fittings and casings, etc., not requiring so much care and attention, might be more generally used for express steamers as they are in naval ships. It is, however, imperative for success in either service that the tubes last at least as long as those of a cylindrical boiler; that they be straight, or nearly so, and inclined at a good angle with the horizon—say, at least, 10 degrees; that the circulation be certain, uniform, and rapid; that the surfaces exposed to heat are to be easily cleaned and kept clean; that the fittings be as few in number as possible and absolutely tight; that the casings be so designed as to remain air-tight and to keep good as long as the boiler itself, and that the grates be of such a size and so placed as to get good combustion as well as uniform distribution of the heat produced either by coal or oil.

CHAPTER XXIV.

BOILERS—CONSTRUCTION AND DETAIL.

Boilers for the mercantile marine are nearly always constructed in accordance with the rules prescribed for the scantlings by the Board of Trade, Lloyd's or other Registry. If a ship requires a passenger certificate, the boiler must be built in accordance with the Board of Trade rules and regulations; and if to class at Lloyd's, those of Lloyd's Registry must be followed. If both a Board of Trade certificate and a Lloyd's machinery certificate are requisite, the boiler must be so designed as to accord with the rules of both. It is much to be regretted that these two public institutions do not agree to one set of rules which shall be acceptable to both, and in accordance with the best engineering knowledge and experience of the day. Both sets of rules are based on scientific principles and practical experiments on a large scale, and differ only slightly and that in form from one another in consequence. They chiefly differ on the question of the factor of safety, and the differences on this point arise from the allowance included covertly for wear and tear; the survey in the one case being yearly, and in the other often at longer intervals.

Ships built for mercantile purposes, but neither classed at a Registry nor requiring a passenger certificate, may have boilers constructed *in accordance with the rules* of one or the other institution.

The Testing of Boilers by Hydraulic Pressure was primarily intended to prove their structural strength and local stiffness to withstand the steam pressure at which they were intended to work; incidentally the workmanship was also proved, but as the working pressure in early days was very low, this latter was quite a secondary consideration. As there were practically no recognised rules such as now obtain to guide both the designer and boilermaker, such a test was absolutely necessary to avoid catastrophe. The extent of test pressure beyond the working limit was a matter of a few pounds in early days, and so long as the box boiler was in use the margin never exceeded 35 lbs. per square inch. The old custom of double the working pressure as the measure of the test was not unreasonable, therefore, as the margin above working pressure by such a rule gave all that was necessary and no more. The cylindrical boiler as made to-day in accordance with well-proved rules of the best and tested materials, and its workmanship quite beyond reproach, would command confidence if not tested by water at all, and any leaks when steam was raised would be quite small and insignificant. Nevertheless, it is good for all concerned that there should be a water test of some kind at a pressure above that at which it has to work; but what that margin should be still excites much controversy and not a little heart-burning. It must be always remembered that the cold-water test conditions are quite

TABLE LXVII.—Tests of Boiler Material (Steel).

Part of Boiler.	Admiralty.		Board of Trade.		Lloyd's Register.		British Corporation.		Bureau Veritas.	
	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.
		Per cent.		Per cent.		Per cent.		Per cent.		Per cent.
Shell plates, Tons per sq. in.	27 to 30	20	27 to 35	18	28 to 32	20	28 to 32	20	to 30	20
Plates exposed to flame, ,,	24 to 27	25	26 to 30	20	26 to 30	23	26 to 30	23	to 28	22
Rivet bars, ,,	24 to 27	25	26 to 30	25	26 to 30	25	26 to 30	25	24 to 27	25
Stay bars, ,,	27 to 30	20	27 to 32	20	28 to 32	20	28 to 32	20	to 30	20
Steam-pipe plates, ,,	24 to 27	33	...	...	...	...	...	...	...	...
Corrugated furnaces, ,,	23 to 25	27	...	...	26 to 30	23	26 to 30	23	to 25	25
Tubes, ...	26	27	26 to 30	20	...	...	...	...	22 to 25	22

By the British Standards Committee's Rule, shell plates for boilers are made of steel whose tensile strength is between 28 and 32 tons, and for plates exposed to flame, or which have to be flanged or heat-treated, it is between 26 and 30 tons tensile.

different from those prevailing on service, for then, instead of being at an uniform temperature throughout, and that, too, at or about that at which the structure was rivetted together, the variation between the parts may be considerable, and all much hotter than when constructed, thereby changing the conditions of stress and strain considerably. The furnaces, chambers, and tubes will all be hotter than the shell and its long stays, and therefore should expand more; on the other hand, the considerable length of the stays in the steam space permits of greater actual extension under load, so that the equilibrium is probably maintained at the high as at the low temperature. High temperature affects the structure of the boiler, but the local stresses and strains only are severe, and it is really on getting up steam that faulty or imperfect workmanship is made manifest; it is for that reason many engineers require that steam shall be raised in one at least of a set of boilers before being placed in the ship after being tested by water. A box boiler, as made years ago for working pressures of 25 to 35 lbs. with flat sides and ends, flattened furnaces, internal uptakes, the stays complicated and riveted to the boiler, and, therefore, with no initial tension, made of untested and somewhat uncertain material by means also uncertain, and designed throughout more or less by rule of thumb, required some drastic tests to give it a character, and ample proof of its reliability was absolutely necessary. Moreover, such a boiler, with its deadweighted safety-valves and huge grates, was liable to have an increase of steam pressure on the sudden and unexpected stoppage of the engines far in excess of the working pressure, and might amount to the 25 to 35 lbs., especially when ships had no auxiliary steam-worked feed pump. Altogether, then, the rule for testing boilers to a pressure equal to twice the working pressure was quite a reasonable one, and was never questioned.

To-day, however, everything is different; there is no uncertainty in design or manufacture, the construction is carried out under constant and careful inspection, and the whole of the material is tested and its character determined; moreover there is no possibility of steam accumulating to an extent as to produce a dangerous pressure; the Admiralty, therefore, were fully justified in changing the rule for water tests years ago on the use of Siemens' steel becoming universal, and the working pressure raised to something over 100 lbs. per square inch. The Bureau Veritas and German Government have also acquiesced in the reasonableness of these views, and for a very long time all locomotive engineers have been content with quite a moderate margin over the working pressure as the test of these boilers. The Board of Trade, Lloyd's Register, and the British Corporation, however, still require all new boilers to be tested by water in the presence of their Surveyor to a pressure double that at which they license them to work.*

The Admiralty Rule for all tank boilers is that they be tested to a pressure equal to 90 lbs. in excess of the working pressure, so that if it is 180 lbs. the test will be 270 lbs.

The Bureau Veritas require a test of double the working pressure, so long as that pressure does not exceed 142 lbs. per square inch; where the working pressure is over 142 lbs., the margin need not exceed 142 lbs.; so that if the working pressure is intended to be 180 lbs., the French test is $180 + 142$, or 322 lbs., instead of 360, as required by English rules.

The German Government require the hydraulic test to be twice the working

* The Rule now established by the British Marine Engineering Design and Construction Committee is that, for boilers whose working pressure does not exceed 100 lbs., the hydraulic test is $2 \times$ W P; when over 100 lbs. the test is $1{\cdot}5 \times$ W P $+ 50$ lbs.

pressure when it does not exceed 5 atmospheres; when the working pressure intended is above this (that is, 75 lbs.) the margin need not exceed 5 atmospheres, so that a German boiler for 180 lbs. W.P. will be tested to (180 + 75), or 255 lbs. only. If a boiler requires testing after it has been at work in the ship the pressure need not be more than 50 per cent. over the working pressure, and not exceed by 85 lbs. the W.P.

It will be seen, then, that for a modern boiler made for a working pressure of 180 lbs. per square inch, the Admiralty would test to 270 lbs.; the German Government to 255; the French officials to 322; and the English to 360 lbs. As a consequence of the latter, a boiler can never be made with such light scantlings as the circumstances would warrant, as the stress would be at test so close to the elastic limit of the material as to be dangerous, whereas under German, French, and Admiralty conditions a factor of safety of 4, which is ample, could be indulged in generally.

The Admiralty rules are not so stringent nor so extended as those of the Board of Trade, as the circumstances of naval construction require a little more elasticity.

Boiler Shell, Cylindrical.—This is the simplest and strongest form to withstand internal pressure; because, since a circle is the figure of least perimeter for a given area, there is no tendency to change of form, the metal is strained in one way only—viz., tangentially and in tension.*

The total pressure tending to rupture a cylinder is the part of all the pressures acting at the various points in direction normal to the surface resolved in one direction. This is equivalent to the pressure on the plane through the axis of the cylinder.

Rupture is resisted by the two sections of metal at the sides.

Let D be the diameter of the thin cylinder, and L its length in inches; p the effective pressure per square inch, and t the thickness of metal in fractions of an inch.

Then the pressure tending to burst it $= p \times D \times L$.

The stress per square inch on the metal resisting this is

$$= \frac{p \times D \times L}{L \times 2t} = \frac{p \times D}{2t}.$$

Let T be the ultimate strength of the material in pounds per square inch, and F be the factor of safety deemed advisable, and let $T \div F = f$.

Then the safe working pressure for a boiler shell, or other cylindrical part subject to internal pressure $= \dfrac{2t \times f}{D}$.

This holds good only when there is no joint or other cause of reduction of effective area of plate section.

Since a boiler shell is made of one or more plates connected by riveted joints, the effective area of plate is that part remaining between the rivet holes (neglecting the effect of the friction between the plates).

If the plates are connected by means of a single row of rivets, the average value of the part remaining between the rivet holes is generally 56 per cent. of the whole plate; so that in this case

$$\text{Safe working pressure} = \frac{2t \times f}{D} \times \frac{56}{100} = \frac{t \times f}{D} \times 1{\cdot}12.$$

* For this and the other parts of the boilers, *vide* new Rules established by the British Marine Engineering Design and Construction Committee as set out in the Appendix.

When the plates are connected by two rows of rivets, so that the joint is said to be a double-riveted one, there is on the average 70 per cent. of the plate remaining between the holes; in this case

$$\text{Safe working pressure} = \frac{2\,t \times f}{D} \times \frac{70}{100} = \frac{t \times f}{D} \times 1{\cdot}40.$$

It will be seen that the double-riveted joint is more than 25 per cent. stronger than the single-riveted. This difference is due to the wider spacing of the rivets which can be allowed, in consequence of the increase of rivet area obtained with the two rows.

The value of F depends on the kind of material, and on its quality.

The Board of Trade requires that the steel used in the shells of boilers shall have a minimum strength of 27 tons and a maximum of 32 tons* per square inch, with an elongation of not less than 18 per cent. in 10 inches, and allows 28 tons to be used for calculations. If the plates stand a test of over 28 tons, that higher figure may be taken for calculation purposes. Steel up to 35 tons has been used for boiler shell, and is likely to be so generally. The Board, however, requires a minimum factor of safety of 4·5, while the Admiralty was content with 4·0, which is quite sufficient, considering that the stress on the material is gently applied and remains quite a steady continuous one during work.

Table lxvii. gives the tests of the materials required by the Admiralty Board of Trade, Lloyd's Register, etc.

Riveting.—The longitudinal seams or joints of a boiler shell are made in one of the following ways :—

(1) *Lap joint and single-riveted.*—The plates in this case are lapped one over another, and united by a single row or rivets. This method is only adopted when the plate is thicker than required for mere strength, so that 56 per cent. of it is sufficient to safely withstand the pressure. This generally arises in the case of small boilers and steam receivers, where the diameter is comparatively small, or with low pressures, when the thickness of plate which is sufficient to withstand the pressure would not permit of caulking, nor give the allowance for deterioration.

TABLE LXVIII.—Strength of Various Riveted Joints.

Description of Joint.	Per cent. of Solid Plate.
1. Lap joint, ordinary single-riveting, 1 rivet to each pitch, . .	56
2. ,, ,, double-riveting, 2 rivets to each pitch, . .	66
3. ,, ,, treble-riveting, 3 rivets to each pitch, . .	72
4. ,, special treble-riveting, 4 rivets to each pitch, . .	80
5. Butt joint, double straps, ordinary double, 2 rivets to each pitch, .	75
6. ,, ,, special double, 3 ,, ,, .	81 to 84
7. ,, ,, ordinary treble, 3 ,, ,, .	80 to 82
8. ,, ,, special treble, 5 ,, ,, .	84 to 88
9. ,, ,, ordinary quadruple, 4 rivets to each pitch,	84
10. ,, ,, special quadruple, 9 ,, ,,	92
11. ,, ,, ,, ,, 11 ,, ,,	93 to 94·5

* The standard steel plates for boiler shells have a tensile strength of 28 to 32 tons.

Rivet metal* is always softer than boiler plate, and its resistance to shearing is less than that of the latter to tension, for this reason the area of the rivets should be greater than that of the plate remaining between the rivet holes; but, on the other hand, whereas the plate is subject to reduction by wear, the rivet section is not so affected, and in consequence it might suffice to allow the same area of rivet. The Board of Trade requires the rivet area of section to be $\frac{27}{23}$, or 1·174 times the area of section of plate between the holes for steel whose tensile is 27 tons.

Taking 56 per cent. as the proportion of plate between the holes, p the pitch, and d the diameter of the rivets,

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 56},$$

or

$$\text{Pitch of rivets} = 2{\cdot}273 \times \text{diameter}.$$

Since the area of rivet = 1·174, the portion of the plate between holes,

$$\frac{\pi}{4} d^2 = \frac{27}{23}(p - d)\,t = \frac{56}{100}\,p \times t \times \frac{27}{23},$$

Substituting the value of p as found above,

$$\text{Diameter of rivet} = 1{\cdot}9 \times \text{thickness of plate}.$$

Example.—To find the pitch and diameter of rivets for a single-riveted lap joint, with plates $\frac{1}{2}$ inch thick, strength of joint being 56 per cent.

$$\text{Diameter of rivet} = 1{\cdot}9 \times \tfrac{1}{2}, \text{ or } 0{\cdot}95 \text{ inch}.$$
$$\text{Pitch of rivets} = 2{\cdot}273 \times 0{\cdot}95, \text{ or } 2{\cdot}16 \text{ inches}.$$

The lap of the plates is three times the diameter of the rivet; if it is made more than this, the plate will spring in caulking; if made less, there is danger of bulging or cracking the edges, and also there is no margin for recaulking if required.

The following table gives the pitch, etc., as found in general practice :—

TABLE LXIX.—Lap Joints, Single-Riveted, Tight Work.

Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.	Breadth of Lap.	Percentage of		Board of Trade P. C. Joint.
				Plate.	Rivet	
$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{3}{16}$	$1\frac{1}{2}$	58·0	66·1	56·2
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{7}{8}$	58·3	65·4	56·0
$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{4}$	57·1	67·3	57·3
$\frac{7}{16}$	$\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{5}{8}$	58·8	65·2	55·9
$\frac{1}{2}$	1	$2\frac{1}{4}$	$2\frac{7}{8}$	55·6	70·0	59·6
$\frac{9}{16}$	$1\frac{1}{16}$	$2\frac{1}{2}$	$3\frac{1}{8}$	57·5	63·3	54·0

(2) *Lap joint and double-riveted.*—Here there are two rows of rivets. Sometimes, as in shipbuilding, the rivets of one row are exactly in line with those of the other, and thus called "chain" riveting; but more frequently those of one row are in line with the middle of the spaces of the other, and it is then called "zigzag" riveting. The latter plan requires less lap, and

* Steel parts for rivets and screwed stays have an ultimate tensile of 26 to 30 tons and its resistance to shear is returned as 23 tons.

makes tighter work than the former, but does not leave the plate so strong, especially when the holes are punched. Here again the rivet area may be made the same as the area of the plates between the holes, but the Board of Trade requires the area of rivet section to be $\frac{27}{23}$ or 1·174 that of the plate between the rivet holes. Taking the proportion of the plate between the holes at 70 per cent.

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 70},$$

or,

$$\text{Pitch of rivets} = 3{\cdot}333 \times \text{diameter}.$$

Also.

$$2 \times \frac{\pi\, d^2}{4} = \frac{70}{100}\, p \times t \times \frac{27}{23} = 0{\cdot}8\, p \times t.$$

whence, substituting the above value of p,

$$\text{Diameter of rivet} = 1{\cdot}7 \times \text{thickness of plate}.$$

Example.—To find the pitch and diameter of the rivets for a lap joint, double-riveted; the plates being $\frac{3}{4}$-inch thick, and the strength of joint 70 per cent.

$$\text{Diameter of rivet} = 1{\cdot}7 \times \tfrac{3}{4}, \text{ or } 1{\cdot}275 \text{ inches.}$$
$$\text{Pitch of rivet} = 3{\cdot}333 \times 1{\cdot}275, \text{ or } 4{\cdot}35 \text{ inches.}$$

Also, *if treble riveted,* and the strength of joint 72 per cent.

$$\text{Pitch of rivets} = 3{\cdot}57 \times \text{diameter}.$$

$$3\,\frac{\pi\, d^2}{4} = \frac{72}{100}\, p \times t \times \frac{27}{23} = 0{\cdot}845\, p \times t,$$

whence, substituting the above value of p,

$$\text{Diameter of rivet} = 1{\cdot}28 \times \text{thickness of plate}.$$

Example.—To find the pitch and diameter of the rivets for a treble-riveted lap joint, the plates being 1 inch thick.

$$\text{Diameter of rivets} = 1 \times 1{\cdot}28, \text{ or } 1{\cdot}28 \text{ inches.}$$
$$\text{Pitch of rivets} = 3{\cdot}57 \times 1{\cdot}28, \text{ or } 4\tfrac{1}{2} \text{ inches.}$$

A Special Arrangement of Treble Riveting is sometimes resorted to whereby the strength of joint is 80 per cent. of the solid plate, and has double the number of rivets in the middle row than in either of the two outer. In this case it is obvious that the pitch in the outer rows must be such that inner row rivets must be far enough apart to give at least 56 per cent. of plate between them. Table lxxii. gives the diameter, pitch, etc., for such a type of joint, and it will be seen that the ratio of d to t is high—viz., 1·50.

General Rules for Riveted Joints are based on the allowances made by the Board of Trade for mild steel—viz., that the shell plates may vary in tensile strength from 27 to 36 tons, and that the ultimate shearing resistance of rivet steel is 23 tons per square inch. The minimum tensile for the plates of 27 tons is assumed in the rules and calculations, and by referring to Table lxix. the adjustment can be made when steel of higher minimum tensile strength is intended to be used.

For lap joints generally, where p is the pitch of outer rows, and d the diameter of rivet—

Area of rivet × number of rivets per pitch × 23 = $(p - d) \times t \times 27$.

(*a*) Then thickness of plate $= \dfrac{d^2 \times n}{p - d} \times 0{\cdot}669$.

For a single-riveted joint, $p - d$ is usually 56 per cent. of p.
,, double- ,, $p - d$,, 66 ,, ,,
,, treble- ,, $p - d$,, 72 ,, ,,
,, special ,, $p - d$,, 80 ,, ,,

Substituting these values in equation (*a*), the following rules hold good :—

Thickness of plates when lap-jointed	$= f \times d^2 \div p$.
For ordinary single-riveting	$= f = 1{\cdot}195$.
,, double-riveting	$= f = 2{\cdot}027$.
,, treble-riveting (3 rivets per pitch)	$= f = 2{\cdot}788$.
For special treble-riveting (4 rivets per pitch)	$= f = 3{\cdot}345$.

Also it follows that :—

For ordinary single-riveting	$d = 0{\cdot}44\,p$;	and $p = 2{\cdot}273\,d$.
,, double- ,,	$d = 0{\cdot}34\,p$;	and $p = 2{\cdot}94\,d$.
,, treble- ,,	$d = 0{\cdot}28\,p$;	and $p = 3{\cdot}57\,d$.
For special treble-riveting	$d = 0{\cdot}20\,p$;	and $p = 5{\cdot}00\,d$.

Example.—A plate $\frac{3}{4}$ inch thick if connected to another by a lap joint will have :—

For single-riveting,	$1\frac{7}{16}$	inches diameter rivets,	2·98	inches pitch.
,, double- ,,	$1\frac{1}{8}$	,, ,,	3·30	,,
,, treble- ,,	1	,, ,,	3·57	,,
,, special ,,	$1\frac{1}{8}$	,, ,,	5·625	,,

For Butt Joints and Double Straps the Board of Trade permits only 1·875 times the single shear instead of the double. Here area of rivet section × number per pitch of outer rows × 23 × 1·875 = $(p - d) \times t \times 27$.

(*b*) That is, the thickness of plate $= \dfrac{d^2 \times n}{p - d} \times 1{\cdot}2543$.

Single-riveting is seldom or never adopted with butt joints.

For ordinary double-riveting,	$(p - d)$	is usually	$= 0{\cdot}75 \times p$.
,, treble- ,,	$(p - d)$	,,	$= 0{\cdot}80 \times p$.
For special 3-rivet double-riveting	$(p - d)$	,,	$= 0{\cdot}81 \times p$.
,, 5-rivet treble-riveting	$(p - d)$	,,	$= 0{\cdot}84 \times p$.
For ordinary 4-rivet quadruple-riveting	$(p - d)$	,,	$= 0{\cdot}84 \times p$.
For special 9-rivet ,,	$(p - d)$	,,	$= 0{\cdot}92 \times p$.
,, 11-rivet ,,	$(p - d)$	,,	$= 0{\cdot}94 \times p$.

Substituting these values in equation (*b*), then :—

Thickness of plates when double-strapped butt-jointed	$= d^2 \times K \div p$.
For ordinary double-riveted joint,	K = 3·333.
,, special ,, ,,	K = 4·652.
,, ordinary treble-riveted joint,	K = 4·704.
,, special treble-riveted joint,	K = 7·466.
,, ordinary quadruple-riveted joint,	K = 5·973
,, special 9-rivet ,, ,,	K = 12·26.
,, ,, 11-rivet ,, ,,	K = 14·68.

In case of the ordinary double-riveted,		$d = 0{\cdot}25\,p$, or $p = 4{\cdot}00\,d$.
,,	special double-riveted,	$d = 0{\cdot}19\,p$, or $p = 5{\cdot}26\,d$.
,,	ordinary treble-riveted,	$d = 0{\cdot}20\,p$, or $p = 5{\cdot}00\,d$.
,,	special treble-riveted,	$d = 0{\cdot}16\,p$, or $p = 6{\cdot}25\,d$.
,,	ordinary quadruple-riveted,	$d = 0{\cdot}16\,p$, or $p = 6{\cdot}25\,d$.
,,	special 9-rivet quadruple-riveted,	$d = 0{\cdot}08\,p$, or $p = 12{\cdot}50\,d$.
,,	special 11-rivet quadruple-riveted,	$d = 0{\cdot}06\,p$, or $p = 16{\cdot}67\,d$.

This Special Form of Treble-riveted Butt Joint now in common use for the longitudinal joints of boiler shell plates, there are three rows of rivets on each side of the joint; and the number of rivets in the outer row, or that next the edge of the butt straps of each, has half the number of rivets that is in each inner row. That means that, instead of six rivets acting to each (outer) pitch, there are only five; or, to make the comparison, there are only $2\frac{1}{2}$ for each inner or general pitch; therefore, if p is taken as the pitch in outer row, n is 5, and by substituting the value of p from $p - d = 0{\cdot}84\,p$,

$$p = 6{\cdot}25\,d.$$

(c) That is, $\quad$ thickness of plate $= \dfrac{d^2 \times 5}{p - d} \times 1{\cdot}1707$; or $\dfrac{d^2}{p} \times 6{\cdot}97$.

Substituting the value of $p = 6{\cdot}25\,d$, then

$$\text{Diameter of rivet} = 0{\cdot}837 \times \text{thickness of plate}.$$

Hence for an inch plate with this style of joint the rivets would be $\frac{7}{8}$, and the pitch $5\frac{1}{2}$ inches.

The advantage of this kind of joint is perceptible for thick plates, as may be seen by calculating the size and pitch of rivets for a plate, say, $1\frac{1}{2}$ inches thick.

Then the diameter is 1·5 and the pitch 9·4 inches, whereas if with the ordinary treble riveting the diameter would be as much as $1\frac{3}{4}$ inches, and the pitch for all the rows $8\frac{3}{4}$ inches, and the strength of the joint 5 per cent. weaker, and not as capable of remaining tight under steam as the other joint with its inner rows of rivets pitched no more than $4\frac{1}{4}$ inches.

The thickness of the butt straps should vary with the pitch of outer rows to resist forcing when being caulked. In practice, the thickness of inner strap should be not less than one-eighth the pitch of the outer rows, so that for the 1·5-inch plate the inner strap should be at least $1\frac{3}{16}$.

The Thickness of Butt Straps required by the Board of Trade is five-eighths the thickness of plate for each of two straps; or nine-eighths of one only for ordinary styles of riveting; with the special one, as above described—

$$\text{Thickness of butt straps} = \frac{5}{8}t \times \frac{p - d}{p - 2\,d}.$$

For the 1·5-inch plate with the pitch as above

$$\text{The thickness required by B. of T.} = \frac{5}{8} \times 1{\cdot}5 \times \frac{9{\cdot}4 - 1{\cdot}5}{9{\cdot}4 - 3} = 1{\cdot}158 \text{ inches.}$$

Breadth of Butt Straps for double riveting is $2{\cdot}5 \times$ pitch; for treble riveting, $3 \times$ pitch; and for treble riveting with the alternate rivets in outer rows omitted, $2{\cdot}45 \times$ the pitch.

Breadth of lap joints single-riveted is $3 \times$ diameter of rivet.
,, ,, double- ,, $3 \times$ diameter $+ 0{\cdot}55 \times$ pitch.
,, ,, treble- ,, $3 \times$ diameter $+ 1{\cdot}10 \times$ pitch.

Treble-riveted Lap Joints with the number of rivets in the middle row double that in the two outer is a good form of joint, and has similar advantages to those possessed by the double butt straps treated in a similar way. The strength of that joint may be calculated on the basis that there are four rivets in single shear for each unit of pitch.

That is, the thickness of plate $= \dfrac{d^2 \times 4}{p - d} \times 0{\cdot}669.$

The highest strength of joint with reasonable size of rivet is 80 per cent.—that is,

$$p - d = 0{\cdot}80 \times p, \text{ or } p = 5{\cdot}0 \times d.$$

Then thickness of plate $= \dfrac{d^2}{p} \times 3{\cdot}345 = \dfrac{d}{5} \times 3{\cdot}345.$

Then diameter of rivet $= 1{\cdot}49 \times t.$

Example.—If an inch plate is jointed with this type of riveting, the diameter will be $1\frac{1}{2}$ inches, and the pitch $7{\cdot}5$ in the outer rows and $3\frac{3}{4}$ inches in the middle one. The percentage of joint through that row will be

$$\frac{3{\cdot}75 - 1{\cdot}5}{3{\cdot}75} \times 100, \text{ or } 60.$$

This is rather too close for good practice; the middle row, therefore, should always have a pitch not less than $2{\cdot}8 \times$ diameter.

With Double Butt Straps and Double Riveting, with the outer rows having half the number of rivets as in the inner, the following holds good. Here there are three rivets in double shear for each unit of pitch, and assuming that the pitch of the inner row is $2{\cdot}6$, and the outer $5{\cdot}2$ times the diameter,

Then strength of joint $= \dfrac{5{\cdot}2 - 1}{5{\cdot}2}$, or $0{\cdot}808$.

The strength of the riveting $= \dfrac{3 \times 1{\cdot}875 \times 23 \times 0{\cdot}785\, d^2}{5{\cdot}2\, d \times t \times 27}$, or $\dfrac{d}{t}\ 0{\cdot}722$.

If it is 81 per cent., then

$$\frac{d}{t} \times 0{\cdot}722 = 0{\cdot}81, \text{ or } d = 1{\cdot}115 \times t.$$

For example, if an inch plate is connected to another with double butt straps and the outer rows with half the number of rivets in the inner, the rivets will be diameter $= \sqrt{1{\cdot}115 \times 1}$, or $1\frac{1}{16}$; the pitch of the middle row will be $2\frac{3}{4}$, and the outer rows $5\frac{1}{2}$. By the ordinary methods the rivets would be $1\frac{1}{8}$ diameter, and the pitch $4\frac{1}{4}$ inches with 75 per cent. of plate and $72{\cdot}8$ of rivets (B. of T. rules), as against the $80{\cdot}8$ per cent. of each member by the above system. It should be noted, however, that the butt straps in this case must be of such a thickness that together they give at least 81 per cent. of solid plate with only 60 per cent. of the pitch between the rivet holes.

$$\textit{The sum of the thickness of butt straps} = \frac{81}{60} \times t, \text{ or } 1{\cdot}35 \times t.$$

Each strap, therefore, ought to be $0{\cdot}7 \times t$; and if the thickness be taken at pitch $\div$ 8 for the inner strap, then for the above example its thickness would be $5\frac{1}{2} \div 8$, or 0·688 inch. The inner strap for such joints should be $0{\cdot}75 \times t$, and the outer $0{\cdot}675 \times t$

The Board of Trade Rule for the maximum pitch of rivets is as follows, but the limit for any thickness of plate is $10\frac{1}{2}$ inches:—

$$\text{Maximum pitch} = c \times t + 1{\cdot}625 \text{ inches.}$$

$c = 2{\cdot}62$ for lap joints with two rivets in the pitch space.
$c = 3{\cdot}47$ „ „ three „ „
$c = 4{\cdot}14$ „ „ four „ „
$c = 3{\cdot}50$ for butt „ two „ „
$c = 4{\cdot}63$ „ „ three „ „
$c = 5{\cdot}52$ „ „ four „ „
$c = 6{\cdot}00$ „ „ five „ „

TABLE LXX.—Lap Joint, Double Riveting Zigzag. Strength of Joint 66 per cent. of Solid Plate. Breadth of Lap = $4{\cdot}62\,d$.

Thickness of Plate.	Rivets.		Breadth of Lap.	Thickness of Plate.	Rivets.		Breadth of Lap.
	Diameter.	Pitch.			Diameter.	Pitch.	
$\frac{1}{2}$	$\frac{3}{4}$	$2\frac{1}{4}$	$3\frac{1}{2}$	$\frac{7}{8}$	$1\frac{5}{16}$	$3\frac{7}{8}$	6
$\frac{9}{16}$	$\frac{13}{16}$	$2\frac{3}{8}$	$3\frac{13}{16}$	$\frac{15}{16}$	$1\frac{3}{8}$	$4\frac{1}{16}$	$6\frac{1}{4}$
$\frac{5}{8}$	$\frac{15}{16}$	$2\frac{13}{16}$	$4\frac{1}{4}$	1	$1\frac{7}{16}$	$4\frac{1}{4}$	$6\frac{5}{8}$
$\frac{11}{16}$	1	$2\frac{15}{16}$	$4\frac{5}{8}$	$1\frac{1}{16}$	$1\frac{9}{16}$	$4\frac{5}{8}$	$7\frac{1}{4}$
$\frac{3}{4}$	$1\frac{1}{16}$	$3\frac{1}{8}$	$4\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$4\frac{13}{16}$	$7\frac{1}{2}$
$\frac{13}{16}$	$1\frac{3}{16}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{3}{16}$	$1\frac{3}{4}$	$5\frac{3}{16}$	$8\frac{1}{16}$

TABLE LXXI.—Lap Joints, Treble Riveting. Strength of Joint 72 per cent. of Solid Plate.
Breadth of Lap = $3\,d + 1{\cdot}35\,\sqrt{d\,(2\,p + d)} = 6{\cdot}852\,d$.

Thickness of Plate.	Rivets.		Breadth of Lap.	Thickness of Plate.	Rivets.		Breadth of Lap.
	Diameter.	Pitch.			Diameter.	Pitch.	
$\frac{3}{4}$	1	$3\frac{9}{16}$	$6\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{7}{16}$	$5\frac{3}{16}$	10
$\frac{13}{16}$	$1\frac{1}{16}$	$3\frac{13}{16}$	$7\frac{5}{8}$	$1\frac{3}{16}$	$1\frac{9}{16}$	$5\frac{9}{16}$	$10\frac{11}{16}$
$\frac{7}{8}$	$1\frac{1}{8}$	4	8	$1\frac{1}{4}$	$1\frac{5}{8}$	$5\frac{13}{16}$	$11\frac{1}{8}$
$\frac{15}{16}$	$1\frac{1}{4}$	$4\frac{1}{2}$	$8\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{11}{16}$	$6\frac{1}{16}$	$11\frac{5}{8}$
1	$1\frac{5}{16}$	$4\frac{11}{16}$	9	$1\frac{3}{8}$	$1\frac{3}{4}$	$6\frac{1}{4}$	12·0
$1\frac{1}{16}$	$1\frac{3}{8}$	$4\frac{15}{16}$	$9\frac{7}{16}$	$1\frac{7}{16}$	$1\frac{7}{8}$	$6\frac{1}{2}$	$12\frac{5}{8}$

TABLE LXXII.—LAP JOINT, TREBLE RIVETING WITH DOUBLE THE NUMBER OF RIVETS IN MIDDLE ROW THAT OF OUTER ONES. STRENGTH OF JOINT, 80 PER CENT. OF SOLID PLATE. BREADTH OF LAP = 7·4 *d*.

Thickness of Plate.	Rivets.		Breadth of Lap.	Thickness of Plate.	Rivets.		Breadth of Lap.
	Diameter.	Pitch.			Diameter.	Pitch.	
$\frac{3}{4}$	$1\frac{1}{8}$	$5\frac{5}{8}$	$8\frac{3}{8}$	$1\frac{1}{16}$	$1\frac{19}{32}$	8	$11\frac{3}{4}$
$\frac{13}{16}$	$1\frac{7}{32}$	$6\frac{1}{8}$	9	$1\frac{1}{8}$	$1\frac{11}{16}$	$8\frac{1}{2}$	$12\frac{1}{2}$
$\frac{7}{8}$	$1\frac{5}{16}$	$6\frac{9}{16}$	$9\frac{3}{4}$	$1\frac{3}{16}$	$1\frac{23}{32}$	9	13
$\frac{15}{16}$	$1\frac{13}{32}$	7	$10\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$9\frac{3}{8}$	$13\frac{3}{4}$
1	$1\frac{1}{2}$	$7\frac{1}{2}$	11	$1\frac{5}{16}$	$1\frac{15}{16}$	$9\frac{11}{16}$	$14\frac{1}{4}$

N.B.—The pitches are in all cases in this joint in excess of the Board of Trade Rule; for example, by the rule an inch plate may have $5\frac{3}{4}$ pitch only, which means the middle row is only $2\frac{7}{8}$ inches.

The chief difficulty with lap joints is in the working of the corners of the plates where the next strake of plating covers the lap; these corners are hammered or machined to a taper, so as to nearly conform to a circle, and the covering plate is slightly joggled, so as to lie evenly on the deformation caused by the lapping. Most boilermakers now, to ensure a good fit, go to the expense of planing or milling the corners fair, and even to extend the taper part beyond the butt end of the plate itself, so as to cause as little deformation as possible.

Butt Joints with Double Straps and Single-riveted.—This form of joint is not often resorted to, as there are two rows of rivets, and only a shearing area of twice that of the one row of rivets, besides all the expense of double straps, which entail the caulking of four seams; the sole advantage it possesses over the double-riveted lap joint is the absence of smithed corners, and that the plates lie wholly in the circle without deformation; this, however, does not compensate for the extra expense and the liability of leakage from the two extra seams.

The strength of this joint is seldom more than 65 per cent. of the solid plate, as more cannot be obtained without placing the rivets so far apart as to prevent the strap from being caulked tight. If the straps are made of *the same thickness as the plate itself*, 70 per cent. may be obtained. Taking 70 per cent. as the strength of joint, the diameter and pitch of the rivets are the same as given for the double-riveted joint, and the breadth of the strap is six times the diameter of the rivets. For example, if the plate is $\frac{3}{4}$-inch thick, the rivets should be $1\frac{5}{16}$ inches diameter, and $4\frac{3}{4}$ inches pitch, the breadth of strap being $6 \times 1\frac{5}{16}$, or $7\frac{7}{8}$ inches.

Butt Joints with Double Straps and Double-riveted.—This is a very general and deservedly favourite form of joint for thick plates, and when well made gives every satisfaction. There is no necessity for smithing or machining the plates, nor of joggling the covering plate of the next strake, although some boilermakers, to avoid the caulking of the ends of the strap where it butts against the next strake, thin down the end and notch out the covering plate, so as to lap over the strap. This makes a very good joint, but is somewhat expensive, and if the plates are properly fitted, there should be no need of such an elaboration.

For each portion of plate between the holes, there is a rivet area equal to four times the area of section of one rivet; but the Board of Trade used to allow only $3\frac{1}{2}$ times the area of one rivet, but now $3\frac{3}{4}$ times.

Under these circumstances the following rules hold good for a joint equal to 75 per cent. of solid plate:—

$$\frac{\text{Pitch of rivets}}{\text{diameter}} = \frac{100}{100 - 75},$$

or, $$\text{Pitch of rivets} = 4 \times \text{diameter}.$$

Also, since $$3\frac{3}{4} \times \frac{\pi d^2}{4} = \frac{75}{100} p \times t \times \frac{27}{23},$$

$$\text{Diameter of rivets} = 1{\cdot}196 \times \text{thickness of plate}.$$

The thickness of each butt-strap must be at least $\frac{5}{8}$ that of the plate, and when the strap between contiguous strakes is simply butted against them, it is better to be of the same thickness as the plate.

Example.—To find the diameter, pitch of rivets, and thickness of straps for a butt-joint double-riveted, and equal to 75 per cent. of solid plate, whose thickness is 1 inch,

$$\text{Diameter of rivets} = 1 \times 1{\cdot}196, \text{ or } 1{\cdot}196 \text{ inches.}$$
$$\text{Pitch of rivets} = 4 \times 1{\cdot}196, \text{ or } 4\tfrac{13}{16} \text{ inches.}$$

Thickness of long strap $\frac{5}{8}$ inch, and of short strap 1 inch if butted, and $\frac{5}{8}$ inch if fitted under the covering plates.

Butt Joints with Double Straps Treble-riveted.—This form has become a necessity since very thick plates have been used for boiler-shells, in order to get adequate sectional area of rivet; it also admits for the same reason of thinner plates being used when desired, so that when there are only a few butt joints it is really more economical to adopt this form of joint. In this case, for each portion of plate between the holes, there is a rivet area equal to six times the area of section of one rivet; but, as before, the Board of Trade only allow this as $5\frac{5}{8}$. The percentage of plate between the rivets is generally 80 per cent. with this type of joint.

Here the pitch of the rivets is $5 \times$ diameter, and, taking the Board of Trade allowances of 23 tons for shear and 27 for tension, then—

$$5\frac{5}{8} \times \frac{\pi d^2}{4} \times 23 = \frac{80}{100} p \times t \times 27,$$

and diameter of rivet $= 1{\cdot}06 \times$ thickness of plate.

Butt Joints, treble-riveted, with half the number of rivets in the outer than in the inner two rows, are now generally adopted as the best for the longitudinal joints of modern boilers, inasmuch as with rivets of quite comparatively small diameter a stronger joint is obtained; in practice it is generally 84 per cent. of the solid plate, and may often be even higher. Taking this percentage and the ratio of tension to shear as before, the following holds good. There are here five rivets for each space in the outer rows, and with the allowance of $1\frac{7}{8}$ for double shear, the acting rivet area $= 5 \times 1\frac{7}{8}$, or $9\frac{3}{8}$ that of one rivet. Then pitch of rivets is 6·25 diameters, and

$$9\frac{3}{8} \times \frac{\pi d^2}{4} \times 23 = \frac{84}{100} p \times t \times 27.$$

Therefore, diameter of rivet $= 0{\cdot}806 \times$ thickness of plate.

In actual practise the rivets are somewhat larger than given by this rule and often equal to the thickness.

TABLE LXXIII.—BUTT JOINTS, DOUBLE STRAPS, DOUBLE RIVETING ZIGZAG. STRENGTH OF JOINT, 75 PER CENT. OF SOLID PLATE. BREADTH OF STRAP $= 6\,d + 1{\cdot}3\,\sqrt{d\,(2\,p + d)} = 9{\cdot}9\,d$.

Thickness of Plate.	Rivets.		Inner Strap.		Thickness of Plate.	Rivets.		Inner Strap.	
	Diameter.	Pitch.	Least Thickness.	Breadth.		Diameter.	Pitch.	Least Thickness.	Breadth.
$\frac{5}{8}$	$\frac{12}{16}$	3	$\frac{12}{32}$	$7\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{16}$	$4\frac{1}{4}$	$\frac{17}{32}$	$10\frac{1}{2}$
$\frac{11}{16}$	$\frac{7}{8}$	$3\frac{1}{2}$	$\frac{14}{32}$	$8\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{8}$	$4\frac{1}{2}$	$\frac{18}{32}$	$11\frac{1}{8}$
$\frac{3}{4}$	$\frac{29}{32}$	$3\frac{5}{8}$	$\frac{15}{32}$	9	1	$1\frac{3}{16}$	$4\frac{3}{4}$	$\frac{19}{32}$	$11\frac{3}{4}$
$\frac{13}{16}$	1	4	$\frac{16}{32}$	10	$1\frac{1}{16}$	$1\frac{9}{32}$	$5\frac{1}{8}$	$\frac{21}{32}$	$12\frac{3}{8}$

TABLE LXXIV.—BUTT JOINT WITH DOUBLE STRAP, DOUBLE RIVETED ZIGZAG, WITH ALTERNATE RIVETS IN OUTER ROWS OMITTED. STRENGTH OF JOINT, 81 PER CENT. OF SOLID PLATE. BREADTH OF STRAP $= 10{\cdot}6\,d$.

Thickness of Plate.	Rivets.		Inner Strap.		Thickness of Plate.	Rivets.		Inner Strap.	
	Diameter.	Outer Pitch.	Least Thickness.	Breadth.		Diameter.	Outer Pitch.	Least Thickness.	Breadth.
$\frac{5}{8}$	$\frac{23}{32}$	4	$\frac{16}{32}$	$7\frac{5}{8}$	$\frac{15}{16}$	$1\frac{2}{32}$	$5\frac{1}{2}$	$\frac{22}{32}$	$11\frac{1}{4}$
$\frac{11}{16}$	$\frac{25}{32}$	$4\frac{1}{16}$	$\frac{17}{32}$	$8\frac{1}{4}$	1	$1\frac{4}{32}$	$5\frac{7}{8}$	$\frac{24}{32}$	$11\frac{15}{16}$
$\frac{3}{4}$	$\frac{27}{32}$	$4\frac{13}{32}$	$\frac{18}{32}$	$8\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{6}{32}$	$6\frac{3}{16}$	$\frac{25}{32}$	$12\frac{9}{16}$
$\frac{13}{16}$	$\frac{29}{32}$	$4\frac{11}{16}$	$\frac{19}{32}$	$9\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{8}{32}$	$6\frac{1}{2}$	$\frac{26}{32}$	$13\frac{3}{16}$
$\frac{7}{8}$	$\frac{32}{32}$	$5\frac{3}{16}$	$\frac{21}{32}$	$10\frac{1}{2}$	$1\frac{3}{16}$	$1\frac{11}{32}$	7	$\frac{28}{32}$	$14\frac{7}{16}$

N.B.—By Board of Trade Rules for the above pitches the butt straps should be 33 per cent thicker. That is, what is $\frac{18}{32}$ should be $\frac{24}{32}$ for the Board of Trade requirements.

TABLE LXXV.—BUTT JOINTS, DOUBLE STRAPS, TREBLE RIVETING. STRENGTH OF JOINT, 80 PER CENT. OF SOLID PLATE. BREADTH OF STRAP $= 6\,d + 2{\cdot}7\,\sqrt{d\,(2\,p + d)} = 14{\cdot}9\,d$.

Thickness of Plate.	Rivets.		Inner Strap.		Thickness of Plate.	Rivets.		Inner Strap.	
	Diameter.	Pitch.	Least Thickness.	Breadth.		Diameter.	Pitch.	Least Thickness.	Breadth.
$\frac{3}{4}$	$\frac{13}{16}$	$4\frac{1}{16}$	$\frac{16}{32}$	$12\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{7}{32}$	$6\frac{3}{32}$	$\frac{26}{32}$	$17\frac{5}{8}$
$\frac{13}{16}$	$\frac{7}{8}$	$4\frac{3}{8}$	$\frac{18}{32}$	$13\frac{3}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$6\frac{1}{4}$	$\frac{25}{32}$	$18\frac{5}{8}$
$\frac{7}{8}$	$\frac{15}{16}$	$4\frac{11}{16}$	$\frac{19}{32}$	14	$1\frac{1}{4}$	$1\frac{11}{32}$	$6\frac{3}{4}$	$\frac{27}{32}$	$19\frac{7}{8}$
$\frac{15}{16}$	1	5	$\frac{20}{32}$	$14\frac{7}{8}$	$1\frac{5}{16}$	$1\frac{13}{32}$	$7\frac{1}{16}$	$\frac{29}{32}$	$20\frac{7}{8}$
1	$1\frac{1}{16}$	$5\frac{5}{16}$	$\frac{21}{32}$	$15\frac{13}{16}$	$1\frac{3}{8}$	$1\frac{15}{32}$	$7\frac{3}{8}$	$\frac{30}{32}$	22
$1\frac{1}{16}$	$1\frac{1}{8}$	$5\frac{5}{8}$	$\frac{23}{32}$	$16\frac{3}{4}$	$1\frac{7}{16}$	$1\frac{17}{32}$	$7\frac{11}{16}$	$\frac{31}{32}$	23

TABLE LXXVI.—BUTT JOINTS, DOUBLE STRAPS, TREBLE RIVETING WITH ALTERNATE RIVETS IN OUTER ROWS OMITTED. STRENGTH OF JOINT, 84 PER CENT. OF SOLID PLATE. BREADTH OF STRAP $= 6\,d + 2{\cdot}5\,\sqrt{d\,(2\,p + d)} = 15{\cdot}2\,d$.

Thickness of Plate.	Rivets.		Inner Strap.		Thickness of Plate.	Rivets.		Inner Strap.	
	Diameter.	Pitch.	Least Thickness.	Breadth.		Diameter.	Pitch.	Least Thickness.	Breadth.
1	$\frac{7}{8}$	$5\frac{1}{2}$	$\frac{22}{32}$	$13\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{11}{32}$	$8\frac{7}{16}$	$\frac{34}{32}$	$20\frac{3}{8}$
$1\frac{1}{16}$	$\frac{31}{32}$	$6\frac{9}{16}$	$\frac{25}{32}$	$14\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{6}{16}$	$8\frac{13}{16}$	$\frac{35}{32}$	21
$1\frac{1}{8}$	$1\frac{1}{32}$	$6\frac{7}{16}$	$\frac{26}{32}$	$15\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{7}{16}$	$8\frac{15}{16}$	$\frac{36}{32}$	$21\frac{7}{8}$
$1\frac{3}{16}$	$1\frac{1}{16}$	$6\frac{5}{8}$	$\frac{27}{32}$	$16\frac{1}{8}$	$1\frac{11}{16}$	$1\frac{1}{2}$	$9\frac{3}{8}$	$\frac{38}{32}$	$22\frac{13}{16}$
$1\frac{1}{4}$	$1\frac{1}{8}$	7	$\frac{28}{32}$	$17\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{17}{32}$	$9\frac{9}{16}$	$\frac{39}{32}$	$23\frac{5}{16}$
$1\frac{5}{16}$	$1\frac{3}{16}$	$7\frac{7}{16}$	$\frac{30}{32}$	$18\frac{1}{16}$	$1\frac{13}{16}$	$1\frac{9}{16}$	$9\frac{13}{16}$	$\frac{40}{32}$	$23\frac{3}{4}$
$1\frac{3}{8}$	$1\frac{1}{4}$	$7\frac{13}{16}$	$\frac{32}{32}$	19	$1\frac{7}{8}$	$1\frac{5}{8}$	$10\frac{3}{16}$	$\frac{41}{32}$	$24\frac{5}{8}$
$1\frac{7}{16}$	$1\frac{5}{16}$	$8\frac{1}{4}$	$\frac{33}{32}$	20	$1\frac{15}{16}$	$1\frac{21}{32}$	*$10\frac{3}{8}$	$\frac{42}{32}$	$25\frac{1}{8}$

Treble-riveted (zigzag) Butt Joint, having half the rivets omitted in the outer rows, and the outer butt strap only covering all three rows; the inner strap covering the inner rows and permitting of heavy caulking. The pitch of the outer row is p, the diameter of rivets d, and the thickness of plate t.

In this case, for each pitch of outer rows there are four rivets exposed to double shear and one to single shear, hence

$$\text{Effective rivet area} = \left(d^2 + 4 \times \frac{15}{8}\,d^2\right) 0{\cdot}7854 = 6{\cdot}676\,d^2.$$

Taking 23 tons as the resistance to shear, and 27 tons to tensile—

$$6{\cdot}676\,d^2 \times 23 = (p - d)\,t \times 27.$$

The strength of this joint should be 85 per cent. of the solid plate, so that $p = 6{\cdot}67\,d$.

$$\text{Then } d = \frac{27}{23} \times \frac{5{\cdot}67}{6{\cdot}696} \times t = t.$$

The pitch of inner rows equal $3{\cdot}333\,d$, and the strength of joint at them

$$= \frac{2{\cdot}333}{3{\cdot}333}, \text{ or 70 per cent.}$$

With such an arrangement of straps and riveting the limit of pitch of $10\frac{1}{2}$ inches will apply only to the inner rows.

Quadruple-riveted (zigzag) Butt Joint, with straps covering all the rows of rivets: p is the pitch of the outer row, that of the two inner rows is one-third p, while in the second row every third rivet is omitted compared with the inner row—that is, for each pitch of the outer row there are two rivets in the next and three in each of the others—hence for each pitch there are nine rivets in double shear. The thickness of the plate is t. Then,

$$\text{Effective rivet area} = 9 \times \frac{15}{8} \times 0{\cdot}7854\,d^2 = 13{\cdot}257\,d^2.$$

* *N.B.*—The Board of Trade Rules do not allow of a greater pitch than $10\frac{1}{2}$ inches for any thickness of plate.

Taking 23 tons as resistance to shear and 27 tons tensile,

$$13{\cdot}257\, d^2 \times 23 = (p - d)\, t \times 27.$$

It is generally convenient to make the rivets of the same diameter as the thickness of plate; then as $d = t$—

$$13{\cdot}257\, d \times 23 = (p - d)\, 27. \qquad \text{Or, } p = 12{\cdot}29\, d.$$

The strength of the joint is then $11{\cdot}29 \div 12{\cdot}29$, or 91·8 per cent.; that of the inner part is $\frac{4{\cdot}1 - 1}{4{\cdot}1}$, or 75·6 per cent.

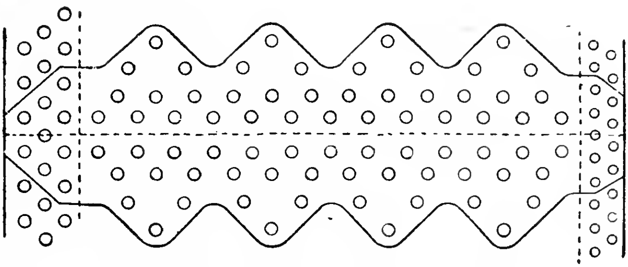


Fig. 237*a*.—Special 9-Rivet Quadruple Joint.

Another Quadruple-riveted Joint has half the number of rivets in the second row than in the two inner, and a quarter the number in the outer row. That is, for each pitch of outer rows there are two rivets in the second and four in each of the two inner rows—that is, there are **11** rivets in double shear per pitch.

The effective rivet area is then $= 11 \times \frac{15}{8} \times 0{\cdot}7854\, d^2$, or $16{\cdot}2\, d^2$.

Then, $$16{\cdot}2\, d^2 \times 23 = (p - d)\, t \times 27.$$

As before, assuming $d = t$, then $16{\cdot}2\, d \times 23 = (p - d)\, 27$. Or, $p = 14{\cdot}8\, d$

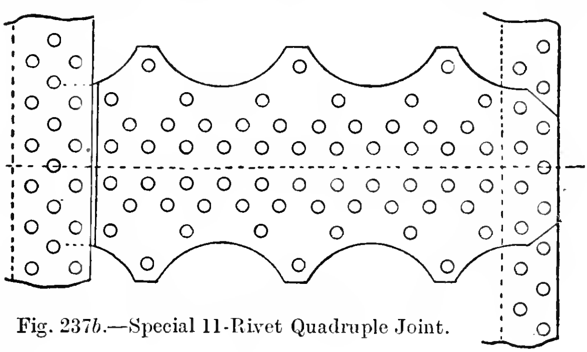


Fig. 237*b*.—Special 11-Rivet Quadruple Joint.

The strength of this joint is then $\frac{14{\cdot}8 - 1}{14{\cdot}8}$, or 93·2 per cent. That through the inner row is $\frac{3{\cdot}93 - 1}{3{\cdot}93}$, or 76·4 per cent., and through the second row $\frac{7{\cdot}4 - 1}{7{\cdot}4}$, or 86·5 per cent.

It is necessary with these quadruple-riveted joints that the butt straps be cut away, following the contour of the riveting into what are called vandykes, in order that the caulking may be effective, and also to save weight.

The following table gives the thickness of shell plates for minimum tensile test strengths from 28 to 38 tons to equal those of 27 tons, for which the tables of pitch and diameter of rivets, etc., are calculated. For example, if a boiler is to be of steel whose miniumm tensile test shall be 36 tons, the riveting suitable is got by finding the thickness of 27-ton steel for such a boiler and working pressures; if it works out to $\frac{3}{4}$ inch it will be seen that if of 27-ton stuff the thickness would be 1 inch; the riveting given in the table for it will be that to employ.

TABLE LXXVII.—Relative Thickness of Boiler Shell Plates for Different Tensile Strengths.

Minimum Tensile Test Strength in Tons per Square Inch.											
27	28	29	30	31	32	33	34	35	36	37	38
0·50	0·483	0·466	0·450	0·486	0·422	0·409	0·397	0·386	0·375	0·365	0·355
0·60	0·579	0·559	0·540	0·524	0·507	0·491	0·476	0·463	0·450	0·438	0·427
0·70	0·675	0·652	0·630	0·611	0·591	0·573	0·551	0·540	0·525	0·511	0·498
0·80	0·772	0·745	0·720	0·699	0·675	0·655	0·635	0·617	0·600	0·584	0·569
0·90	0·868	0·838	0·810	0·784	0·760	0·736	0·714	0·694	0·675	0·657	0·640
1·00	0·965	0·931	0·900	0·873	0·844	0·818	0·793	0·771	0·750	0·730	0·711
1·10	1·061	1·024	0·990	0·961	0·929	0·900	0·872	0·848	0·825	0·803	0·782
1·20	1·157	1·117	1·080	1·048	1·013	0·982	0·925	0·925	0·900	0·876	0·853
1·30	1·254	1·211	1·170	1·135	1·097	1·064	1·031	1·003	0·975	0·949	0·924
1·40	1·350	1·304	1·260	1·223	1·182	1·145	1·110	1·080	1·050	1·022	0·996
1·50	1·447	1·397	1·350	1·310	1·266	1·227	1·190	1·157	1·125	1·095	1·067
1·60	1·543	1·490	1·440	1·397	1·351	1·309	1·269	1·234	1·200	1·168	1·138
1·70	1·640	1·583	1·530	1·484	1·435	1·391	1·348	1·311	1·275	1·241	1·209
1·80	1·736	1·676	1·620	1·572	1·519	1·473	1·428	1·388	1·350	1·314	1·280
1·90	1·833	1·769	1·710	1·659	1·604	1·554	1·507	1·465	1·425	1·387	1·350
2·00	1·929	1·862	1·800	1·746	1·688	1·636	1·586	1·542	1·500	1·460	1·421

Circumferential Seams* are almost always lapped and double-riveted. Boilers of small size and for working pressures under 100 lbs. are single-riveted, and as one row of rivets is quite sufficient to ensure tightness of joint, which is all that is required in this joint, the single-riveted joint answers the purpose. Boilers made for very high pressures are often treble-riveted in the middle circumferential seams, the Board of Trade rules admitting of a thinner plate when this is done. In the case of large double-ended boilers, the circumferential seams should be always treble-riveted except the end ones, which seldom need be done in this way, and then only when the plates are very thick.

Methods of Work.—Formerly most of the holes in boiler plates were punched and drifted fair. Since the Board of Trade placed a premium on drilled holes, machines have been made which compete successfully, both in the cost and speed of output, with the punching machine, so that now all the holes in a boiler are drilled, and generally done in place.

* To maintain a strength of joint of 60 per cent. for single-ended and 64 for double-ended with thick plates, treble riveting is necessary to obtain sufficient rivet area.

Material.—The shell of a cylindrical boiler is now made of steel plates with a strength up to 32 tons, while the Admiralty limit it to 30 tons; it is expected to stretch 20 per cent. per fracture (see Table lxvii.). Steel of a higher strength is sometimes used, and that with 35 tons as the limit must have 18 to 20 per cent. extension; sometimes steel as high as 40 tons per square inch has been used for shells; it should have the same extension.

Siemens steel has quite taken the place of iron in boiler-making, and, as it possesses a much higher tensile strength, with greater toughness, it is a more suitable material for the purpose. A boiler made wholly of steel was cheaper than when wholly of iron, which no doubt was one cause of the increased demand for steel boilers. Steel plates can, at a trifling extra expense, be supplied of very large sizes, exceeding largely those made by the old Yorkshire ironmasters. As a matter of fact, the overhead price of a heavy specification of steel plates was not seriously greater than that of a light specification, while there was a very considerable difference for iron if large and heavy plates were included. The size of plates to be used in the construction of the shell depends now on the appliances of the boiler-maker; the breadth of plate is limited by the depth of gap in the riveting machine, and the length of plate by the capabilities of the planing machine and squeezer or rolls; and the weight of the plate is limited to the strength of the various small cranes, etc. Steelmakers can make plates up to 12·5 feet broad and 50 feet long, and $1\frac{5}{8}$ inches thick (Chap. xxx.).

Nowadays rolls or squeezers for curving plates, planing machines for truing the edges, drilling machines for dealing with all the holes, and riveting machines for closing the joints are made in such large sizes that a double-ended boiler of the largest kind can be made in two drums, and each drum in two plates; single-ended boilers in one drum, and each drum in two plates; some people prefer to make each drum in one plate when the diameter permits of it. As a rule, it may be assumed that a single-ended boiler has one drum or strake of plating. the strake consisting of two plates; a double-ended boiler should have three strakes of plating, and the longitudinal seams should be so arranged that no two of them come in line nor interfere with the seams at the ends, and they should be well above the furnace line.

Allowance for Wear.*—All boilers are designed so as to last as long as possible, and, since wear takes place by corrosion, some additional thickness must be provided at first to meet this condition. The Board of Trade tacitly make this allowance by using a high factor of safety; but since the factor of safety causes the additional thickness to be *proportional to the total thickness*, while the wear takes place independently of thickness, it does not properly meet the case. A boiler with plates $\frac{1}{2}$ inch thick will waste the same quantity of steel per square foot as one with plates 1 inch thick if worked under similar conditions. Suppose such waste to be $\frac{1}{8}$ inch in a certain time, the loss in one case is 25 per cent., but only $12\frac{1}{2}$ per cent. in the other. To meet the case properly, the factor of safety should be reduced, and a constant quantity added as is done by the Bureau Veritas; for example, in the case mentioned above the plates should be $\frac{9}{16}$ inch and 1 inch, so that at the end of the time the former should be $12\frac{1}{2}$ per cent. under $\frac{1}{2}$ inch in thickness. If any further proof be needed, it is only necessary to calculate the thickness

* Now that salt water is seldom admitted to a boiler, and greater care is taken, the waste of material is only slight.

of plates for boilers of small diameter and low pressure to find it such as would be impossible to rivet and caulk tight.

The following rules make due allowance for such contingencies:—Let D be the diameter of the shell in inches, p the working pressure in pounds per square inch, F a factor; then

$$\text{Thickness of shell plates} = \frac{D \times p}{F} + 2 \text{ (in 32nds).}$$

TABLE LXXVIII.—Values of F on Condition.

When the Longitudinal Joints are	Tensile of Plates.	
	28 Tons	35 Tons.
1. Lap, ordinary, single-riveting,	562	702
2. ,, ,, double-riveting,	662	827
3. ,, ,, treble-riveting,	719	900
4. ,, special, treble-riveting (4 rivets per pitch),	762	952
5. Butt, double straps, ordinary, double-riveting,	750	938
6. ,, ,, special, 2 rows, 3 rivets per pitch,	812	1,015
7. ,, ,, ordinary, treble-riveting,	812	1,015
8. ,, ,, special, 3 rows, 5 rivets per pitch,	850	1,062
9. ,, ,, quadruple, 4 rows, 9 rivets per pitch,	920	1,150
10. ,, ,, quadruple, 4 rows, 11 rivets per pitch,	941	1,176

D is the internal diameter in inches.
p is the working pressure.

Boiler Ends.—There are several methods of connecting the end-plates to the shell.

(1) *Flanging the end-plates.*—This is now the universal and the best plan, for there are only one set of rivets and two caulking edges, and the room occupied is less than with angles. Flanging is done by special appliances at a trifling cost in various designs of hydraulic press. The flanges (fig. 238) are usually inside the boiler, but when the shell is of so small a size that the rivets cannot be held up inside, or it is desired to rivet the joint by machine, the *front* end is placed (fig. 239) with the flange outside. In either way there is no tendency to force the joint by pressure on the ends, as there is with

(2) *Flanging the shell plates*, which some few engineers, in order that a thicker plate may be provided to withstand the wear that takes place at the bottom corners, or rather edges, of the cylindrical boiler, and to avoid the fitting of the flanged ends into the shell with fair accuracy, resort to turning it inwards (fig. 240), so as to form a connection for the ends. This has some bad features, among which may be cited the difficulty and cost of flanging thick plates across the grain, especially after being bent; the necessity for such shell plates to be of low tenacity to have so much ductility; and lastly, the pressure on the end always tending to open the joints. It has, however,

some few advantages, but they are dearly bought, for, to avoid the chance of leakage at the corners of the end-plates, there is far greater risk of leakage from the horizontal shell joints, which are of necessity flanged over too, unless welded. It may be added that such an arrangement quite prevents the furnaces from being near the shell.

Riveting.—The cross seams of the backs are usually double-riveted, although for pressure under 100 lbs. single-riveting does quite well. The riveting of the ends to the shell is usually of the same design as that of the other circumferential seams, but, in double-ended boilers, they need not be treble-riveted because the middle seams are.

The Quality of Plate depends largely on the amount of flanging to be done. To stand flanging around the edges of the circular plates, they may

Fig. 238.

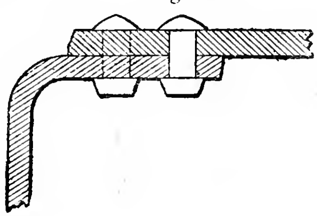
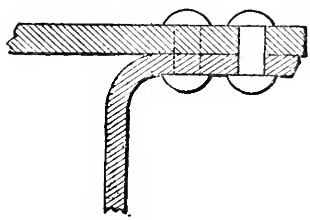


Fig. 239.



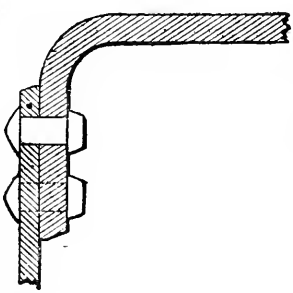


Fig. 240.

Figs. 238 to 240.—Methods of Connecting Shell and End Plates.

be of the same quality as the shell; but, when the furnace holes are flanged to meet the furnaces, a softer quality is desirable, and it is better to use the mild kind as ordered for internal parts, having an extension of 25 per cent. at least to stand such severe treatment.

The Thickness of the End-plates and the pitch of the stays are interdependent to a certain extent; but, since the stays in the upper part of the boiler must be wide enough apart to admit of a man passing between them, the plates at the upper part of the ends must be made thick enough to suit this pitch of the stays, which are usually, in consequence, somewhat thicker than the other part of the ends; some makers, however, prefer to make the ends of one uniform thickness, and stiffen the top plates to stand the wide pitch of the stays by riveting on thick and large washers in wake of these stays, as provided for in the Board of Trade rules. The end-plates are generally from $\frac{5}{8}$ to $\frac{7}{8}$ inch thick, although a few makers prefer to have thicker plates, so as to avoid the necessity for doubling plates about the man- and mud-holes, and for fitting nuts and washers to the screwed stays. Taking 15 inches as the smallest convenient pitch for the stays in the steam space, and suppose them to have riveted washers as well as double nuts, Lloyd's requires the plate to be $\frac{13}{16}$ inch thick for 150 lbs. working pressure, and $\frac{15}{16}$ inch thick for 200 lbs. The Board of Trade permit plates $\frac{1}{32}$ inch less for these pressures.

Furnaces.—Those fitted in the cylindrical boiler are invariably of circular section, that being the best form to resist a uniform external pressure. The strength of such a furnace varies as the square of the thickness of plate, and inversely as the diameter and length. From experiments made on a large scale, there is, however, reason to doubt the supposition that the strength varies *exactly* inversely as the length.

Let D be the external diameter, in inches, of the furnace whose length is L feet, t the thickness of the plates in parts of an inch.

$$\text{Safe working pressure} = \frac{99{,}000 \times t^2}{(L + 1) \times D} \text{ (Board of Trade).}$$

Or,

$$\text{Safe working pressure} = \frac{89{,}600 \times t^2}{L \times D} \text{ (Lloyd's Registry).}$$

When the plain part of a furnace exceeds 120 times the thickness of the plate,

$$\text{Safe working pressure} = \frac{51{\cdot}5}{D}(18{\cdot}75\,t - 12{\cdot}36\,L) \text{ (British Corporation).}$$

If the furnaces are constructed by welding the plates together, or connecting them by a butt joint with double straps single-riveted, or by a single butt strap double-riveted, the Board of Trade allow the above factor. To avoid making by the above rules a furnace which might give way by the crushing of the material, the Board of Trade insist that, in no case shall the working pressure, in pounds per square inch, exceed $\frac{9{,}900 \times t}{D}$, and Lloyd's that the working pressure shall not exceed $50\left(\frac{(300\,t - 12\,L)}{D}\right)$, when the length of the plain part is less than 120 times the thickness of plate.

Rule for Thickness of a Plain Furnace.*—The following gives a suitable thickness for the furnace plates in 32nds of an inch, and makes *due allowance for uniform wear of the surfaces* :—

$$t = \sqrt{\frac{p \times L \times D}{F}} + 2.$$

Here L is the length and D the diameter, ***both in inches;*** and F a factor which is for ordinary mild steel, 1,100.

Furnace plates are made from $\frac{3}{8}$ inch to $\frac{3}{4}$ inch thick, the most general sizes being $\frac{7}{16}$ to $\frac{9}{16}$ inch. Some engineers, however, in spite of this, to avoid using corrugated furnaces or fitting stiffening rings, made the furnace of plates as thick as $\frac{7}{8}$ inch; if the boiler is kept quite clean there is no objection.

If the furnace is so long that $\frac{9}{16}$-inch plates are insufficient by the rules for the working pressure, some means of stiffening it should be resorted to; such stiffening was generally effected by means of rings, as the effective length for purposes of calculation is thus virtually the longest distance between

* For new Rules of the British Marine Engineering Design and Construction Committee, *v.* Appendix.

such rings, or between such rings and the ends, it may be taken as the value of L in the foregoing formulæ.

The Methods of Stiffening Furnaces were—(1) *By making the furnace in two or more drums and connecting them by means of a* U*-shaped hoop*, called the "Bowling hoop" (fig. 241), because first made by the Bowling Company. These hoops are weldless, and possess a considerable amount of elasticity; so that, in addition to stiffening, they allow expansion longitudinally on the part of the furnace. This was a very convenient plan, as it admitted of the furnace being partially withdrawn in case of damage, etc., and notwithstanding that there are two thicknesses of plates, and two laps at each joint of the furnace, it gave every satisfaction when tried. No one, however, makes these hoops now, so that this type of joint has gone out of use.

(2) *By making the furnace in two or more drums, and connecting them by means of flanges*, formed by turning the plate end outwards (fig. 242). To allow of a caulking edge on both sides of the lap, a thin ring is introduced between the flanges. This was a favourite method, because no joint or riveting is exposed to the fire. Such joints generally give trouble from the

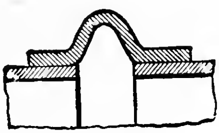


Fig. 241.

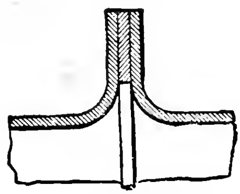


Fig. 242.

Fig. 243.

Figs. 241 to 243.—Methods of Stiffening Furnaces.

strain on the boiler end tending to open the joint, and by the wearing away of the metal at the root of the flanges, due to mechanical action. Furnaces made on this plan also required more room in the boiler, and the flanges blocked up the space between them and the boiler shell.

(3) *By making the furnace in two or more drums of different diameters;* those of small diameter are flanged out so as to fit into, and be connected to, the larger ones with lap joints single-riveted (fig. 243). It possesses one or two very useful features, among which may be reckoned the capability of the furnace, being small at the mouth, to leave good space on the boiler fronts for manholes, etc., and while small at the combustion-chamber ends, it is of good diameter in the middle. It has, however, the objection of presenting joints to the direct action of the fire.

(4) *By making the furnace with a series of corrugations or ridges.*—There are now several ways of accomplishing this, the best known being that of the late Mr. Fox (fig. 244), and that of Mr. Purves (fig. 245), made by John Brown & Co. of Sheffield, the more recent patent being that of Mr.

Morison (fig. 246); this form possesses the good features of the Purves, and avoids the fault of the Fox. Fig. 247 is the modified form of corrugated furnace made by C. D. Holmes. Fig. 248 is the form in which Mr. Deighton, of Leeds, makes furnaces which have proved quite satisfactory.

The corrugated furnace was an extension of the Bowling-hoop principle,

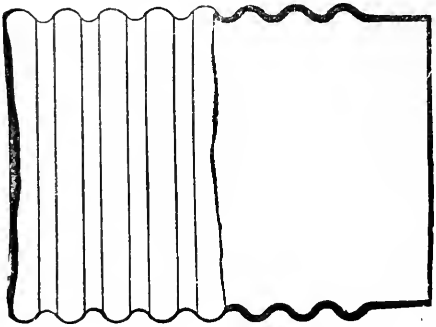


Fig. 244.—Fox's Furnace.

Fig. 245.—Purves' Furnace in Section.

Fig. 246.—Morison's Furnace in Section.

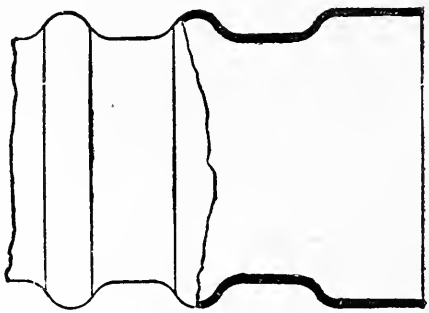


Fig. 247.—Holmes' Furnace.

and its genesis is best illustrated by reference to fig. 247, which shows the plan followed by Mr. Holmes. Here there are comparatively few corrugations, but still sufficient to give the necessary stiffness to the furnace. For increased pressure, such a furnace as this must either be made of thicker

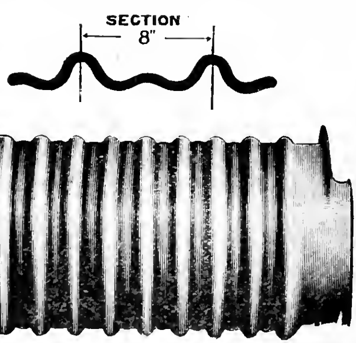


Fig. 248.—Deighton's Furnace.

plate or have the corrugations closer together; consequently, for the same pressure and dimensions, the Fox furnace will be thinner than that of the Holmes. On the other hand, the Holmes furnace is more rigid longitudinally than the Fox furnace. The corrugated furnace made by the Farnley Company has the corrugations formed spirally around the furnace, and are said thereby to give greater longitudinal rigidity, without sacrificing too much of its transverse stiffness; it must, however, tend to set up twisting strains when end-pressure is applied, which would bring shear on the rivets, and the transverse stiffness can be only little more than that of the Holmes furnace.

These special furnaces soon became a necessity for large diameters and high pressures; but, although immensely strong so long as the metal is cold, corrugated ones will probably collapse *longitudinally* when red hot quicker than an ordinary furnace, from the fact of there being superabundance of plate between the extreme points of support to supply the extra length of the arc over that of the chord; a common furnace cannot come down in this way without stretching the metal; in the Fox design the corrugations are simply drawn out of shape.

The Purves furnace was practically an extension of the Adamson-joint principle, and is shown in fig. 245. This furnace possesses quite as much transverse stiffness as the Fox, while being superior in longitudinal rigidity; it is, moreover, easier to clean and to repair.

For corrugated furnaces, the following rules hold good:—

(*a*) $$\text{Working pressure} = \frac{14{,}000 \times \text{thickness in inches}}{D} \text{ (Board of Trade)},$$

D being taken as the smallest outside diameter in inches.

(*b*) $$\text{Working pressure} = \frac{C \times (T - 2)}{D} \text{ (Lloyd's Register)}.$$

T is the thickness in sixteenths of an inch.
D is the smallest diameter outside.
C = 1,259 for Fox's, Morison's, Deighton's, Beardmore's, or Leeds Forge bulb furnaces where made of steel, 26 to 30 tons tensile strength.
C = 1,160 for Purves', when the ribs are 9 inches apart; or Brown's, ribs 8 or 9 inches apart.
C = 945 for Holmes', and C = 912 for Farnley's.

(*c*) $$\text{Working pressure} = \frac{C \times (T - 2)}{D} \text{ (British Corporation)}.$$

C = 1,250 for Leeds Forge bulb furnaces.
C = 1,160 for Fox's, Purves', Deighton's, Morison's, and Brown's.
C = 950 for Holmes' and Farnley's.

(*d*) $$\text{Working pressure} = C\,\frac{(T - 2)}{D} \text{ (Bureau Veritas)}.$$

C = 1,260 for corrugated and bulb furnaces.
C = 1,160 for ribbed furnaces.

Fig. 249 shows a Morison furnace with its back end so formed as to permit of the furnace being withdrawn through the hole in front of the boiler, without sacrificing any of the good features of the method of jointing of the combustion chamber.

Messrs. John Brown & Co. have also a furnace with a back end formed so as to permit of the furnace being withdrawn in the same way. In this case, however, the furnace is reduced somewhat in diameter and flanged outward all round (*v.* fig. 251*a*). This flange is not concentric with the body of the furnace, its centre being higher so as to permit of the connection of the tube plate in the usual way, the back tube plate being flanged and riveted into the combustion chamber all round, and formed with a round hole corresponding to that of the furnace neck.

* For new Rules of the British Marine Engineering Design and Construction Committee, *v* Appendix.

Methods of Connecting Furnaces to End-plates.—There are two distinct ways of accomplishing this:—

(1) *By flanging the furnace to meet the front plate.*—This was formerly a common one, because the iron of a furnace was always of a soft high quality, capable of being easily flanged, but the method is objected to as the pressure on the ends tends to open the joint. The flanging is invariably outwards, and when done the root should have good curvature, the radius of the outer surface being at least $1\frac{1}{2}$ inches. Since steel has been used for boiler ends this method has been practically dropped, as it does not admit of the furnace being withdrawn in case of need.

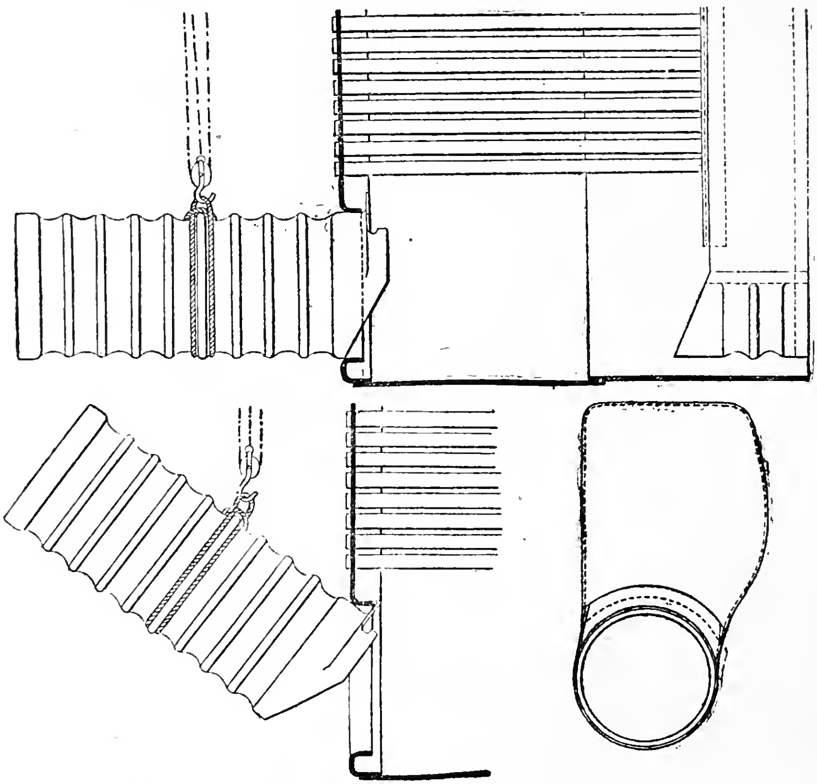


Fig. 249.—Removable Furnace by the Leeds Forge (Ashlin's Patent).

(2) *By flanging the front plate* inwards (fig. 250) or outwards (fig. 251) to meet the furnace. The former is the most general method, the latter being resorted to only when the boiler is so small as to prevent the rivets from being properly "held up," and when it is necessary to have the furnaces very near the shell. In naval ships the boilers were often made in this way. This flanging in or out makes the best finish to the boiler front, and leaves more room for man- and mud-holes, etc., and permits of much neater furnace fronts. The joint (fig. 250) is capable of being caulked at both edges, and

the strain on it is across the rivets, and does not, in consequence, tend to start the caulking.

Combustion Chambers.—The length of a combustion chamber, measured in line with the furnace, should be such that its capacity above the level of the fire-bars is equal to the total capacity of the furnace, when the boiler is single-ended; when double-ended and one combustion chamber is common to opposite furnaces, the capacity of the combustion chamber should be equal to three-fourths of the combined total capacity of the two furnaces.

To obtain such a capacity of combustion chamber when the boiler is single-ended, or double-ended and divided transversely, the length must be about two-thirds the diameter of the furnace, and when common to two opposite furnaces, it must nearly equal the diameter of furnaces.

Combustion chambers are generally formed with flat tops, but made sometimes by curving the back plate over the top to meet the flange of the tube plate. The latter plan avoided the necessity of the girder stays to support the flat top, and reduced the number of joints of plating,

Fig. 250. Fig. 250*a*.

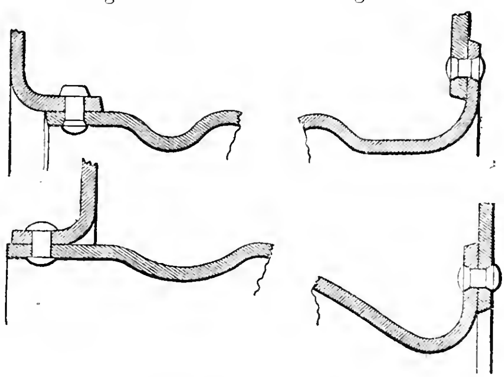


Fig. 251. Fig. 251*a*.

Figs. 250 to 251*a*.—Methods of Connecting Furnaces to Boiler Ends and Tube Plates.

but the capacity of the combustion chamber is less, and the space for tubing, etc., contracted. It used to be claimed for this form that staying is avoided, but this is not a substantial gain, as, in a single-ended boiler, the stays which are necessary for the back end-plates form the stays of the chamber, and in a double-ended boiler, if stays are omitted between the chambers, the Board of Trade surveyors require additional staying in the steam space *to tie the ends of the boiler together.* Although the plan was a favourite one with engineers a few years ago, when the pressure was under 100 lbs., it is now seldom seen.

The thickness of plates and pitch of stays are, of course, interdependent, but, as a rule, the chambers of large boilers, whose working pressure is 150 lbs. per square inch and upwards, are made of $\frac{1}{2}$-inch to $\frac{3}{4}$-inch plates, and those of smaller boilers, or those working at lower pressures, are made of $\frac{7}{16}$-inch to $\frac{1}{2}$-inch plates.

By the Board of Trade rules, a stay-bar $1\frac{3}{8}$ inch in diameter, screwed 10 threads per inch, will sustain 73 square inches at a working pressure of 150 lbs., and only 55 square inches at 200 lbs., while a $1\frac{5}{8}$-inch stay, with 9 threads per inch, will sustain 77 square inches at a pressure of 200 lbs. A plate $\frac{1}{2}$ inch thick requires a stay for 7·74 inches pitch, or 60 square inches at a pressure of 150 lbs., while a $\frac{5}{8}$-inch plate requires one for 9·3 inches pitch or 86 square inches for that pressure; for 200 lbs. pressure, the $\frac{5}{8}$-inch plate requires a stay for a pitch of 8 inches or 66 square inches.

Board of Trade Rule :—Pressure $= \dfrac{C \times (T + 1)^2}{S - 6}$.

T is the thickness of flat plate in sixteenths of an inch; S, area of surface supported in square inches; C, a constant which, for screwed stays with nuts and plates exposed to flame, but in contact with water, is 100.

For stays in steam space fitted with riveted washers, two-thirds the pitch in diameter, nuts, etc., 210 if washers are of the same thickness as plate, but only 165 if with plain washers two-thirds the thickness of plate and three times the diameter of stay.

The bottom of the chambers should be $\frac{1}{16}$ to $\frac{1}{8}$ inch thicker than the sides, as from various causes there is often rapid wear in that part; also to avoid excessive staying and to provide for burning, which sometimes takes place there, the plates at the top should be $\frac{1}{16}$ inch thicker.

Some steel makers now supply plates having a varying thickness so that one single plate can be wrapped around the combustion chamber so as to form the top, sides, and bottom, with the latter of the necessary extra thickness.

The back tube plates vary in thickness from $\frac{9}{16}$ in small boilers for low pressures, to $\frac{3}{4}$ inch, and even $\frac{7}{8}$ inch, in large ones for high pressures. Generally, in modern boilers of ordinary sizes and pressures, the back tube plate is $\frac{5}{8}$ to $\frac{3}{4}$ inch thick, the former being the best size when possible, as with it the tubes can be made quite tight, and there is less liability of cracking the plates or burning the tube ends than with the thicker plates.

The back plate and tube plate of the combustion chamber are almost invariably flanged inwards to take the side plates and those on the top and bottom; some makers have tried to make the chambers by flanging the sides top and bottom to meet the back and tube plates (*v.* fig. 203); but, as this is very troublesome to effect, and prevents the tubes from being extended to the sides and top, it is seldom followed. The flange of the top plate of the furnace should be inside the chamber, and connected to the tube plate with counter-sunk rivets; the landing edge is then turned away from the "wash" of the flame, and no rivet heads are exposed to it. The landing edges of all joints of plating exposed to water should be downwards, so that deposit cannot lodge on them; when they are upwards on the water side the deposit on them is very considerable, and it is found that rapid corrosion then takes place in the angle beneath it. The back tube plate is by some engineers welded to the furnace, so that no joint is exposed to the severe action of the flame issuing from the fuel. It is an expensive thing to do, and utterly prevents the withdrawal of a damaged furnace when done.

Tubes.—The tubes in the ordinary marine boiler are from $2\frac{1}{2}$ inches to 4 inches external diameter, the usual sizes being from $2\frac{1}{2}$ inches to $3\frac{1}{2}$ inches in the mercantile marine, and $2\frac{1}{2}$ inches to $2\frac{3}{4}$ inches in H.M. Navy. With the ordinary natural draught the tubes should not be more than 24 diameters long; with the forced draught they may be as much as 60 diameters long, as in a locomotive boiler, but the practice in torpedo boats and steam launches was about 35 diameters long, and generally with Howden's forced draught is 30 to 36. The length, however, does not matter so much with forced draught.

The spacing of the tubes often depends on circumstances, but in the mercantile marine, where space and weight of machinery are not of such moment as in the Navy, the pitch of the tubes is usually $1 \cdot 4 \times$ diameter. There is less liability to prime when the tubes are widely spaced, and they are more easily cleaned from scale. For the latter purpose they are arranged in rows, both horizontally and vertically, and not zigzag, as often seen in locomotive boilers.

Tubes are manufactured of a certain minimum thickness, and said to be "according to list" when so made. If the pressure they are intended to withstand does not exceed 60 lbs., they may be "according to list"; if it does not exceed 100 lbs., they should be "1 gauge thicker than the list"; if the pressure does not exceed 150 lbs., the tubes should be "2 to 3 gauges thicker than the list." By rule, L.S.C. $= 300 \div d \times \sqrt{p}$.

The following table gives the thickness under the various circumstances in the numbers of the Legal Standard Gauge:—

TABLE LXXIX.—Boiler Tubes.

External diameter of tubes, Ins.,	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	4
Thickness for 60 lbs., - L.S.G.,	12	12	11	11	11	10	10	10	9
Thickness to 100 lbs., - L.S.G.,	11	11	10	10	10	9	9	9	8
Thickness to 150 lbs., - L.S.G.,	10	10	9	9	8	8	7	7	6
Thickness to 200 lbs., - L.S.G.,	9	9	9	8	8	7	7	6	6

It is usual to make the tubes of slightly larger ($\frac{1}{16}$ to $\frac{1}{8}$ inch) diameter at their front end, so as to draw out easily when once started from the plates. Tube manufacturers will swell the ends to $\frac{1}{16}$ inch larger diameter without extra charge.

Boiler tubes were formerly of iron, hard copper, or brass; now they are exclusively of iron or steel; when of the former they are made from strips of best Staffordshire, or other good iron having a tensile strength of 20 tons with an elongation of 12 per cent. When of steel, they are made either of strips having a tensile not exceeding 27 tons, or solid drawn from mild steel billets. Those for water-tube boilers are always solid drawn, cold finished, and carefully made to gauge thickness and character.

Iron tubes are generally used in the mercantile marine; brass tubes were used in the Navy, partly because of their superior conducting power, but chiefly on account of their endurance and reliability, as the iron tubes then made seldom lasted more than four years, and after only as many days a few would sometimes prove defective from small holes being formed, which rapidly enlarged with the rush of water and steam through them.

Brass tubes were made of a composition of 68 per cent. of B.S. copper and 32 per cent. of spelter, which, when drawn out into the tube, has a very high tensile strength and is very tough. Such tubes lasted ten or twelve years under ordinary circumstances, but, if used with coal containing much sulphur, they perished more rapidly and lost the toughness. They cost about four times the price of iron tubes, but, when condemned, were worth about half the original cost, and, since they lasted at least double the time and their superior efficiency was a sufficient set-off for interest on capital, the brass tubes were more economical than the iron ones.

Steel tubes are now being used on a large scale; the early experience with this metal for tubes was not always happy; there is, however, steel and steel, and because tubes of that material failed egregiously many years ago, it is no reason why tubes made of modern steel should not be used now, especially in steel boilers. The Admiralty use welded steel tubes in cylindrical boilers, and the tube-makers are confident that they can make as perfect a weld with the mild steel strips they use as with iron. For internal pressure, however, a solid-drawn tube is preferred, and always used.

To preserve the ends of the tubes in the back tube plate from being wasted away and the severe action of flame on the tube plates with forced draught, the Admiralty used iron ferrules fitted to them. These ferrules (fig. 252) were made of malleable cast iron; they have a slight taper and a mushroom-shaped flange, so that when driven home they are very tight and protect the end of the tube and surrounding plate completely.

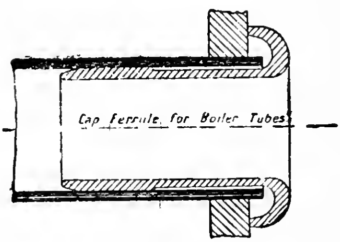


Fig. 252.

Stay Tubes. — The tube plates are usually held and stiffened by some of the tubes of greater thickness being arranged as stays; for that purpose they are screwed at the ends with a fine thread (9 to 10 threads per inch), and either tapped into both plates or tapped into the back plate and screwed by nuts to the front plate. The better plan is the former, the back end thread is *minus* and the front end *plus*, and the tube screwed into both plates at the same time. The section at thread in this plan is in excess of that of the tube owing to its large diameter, and the tube can be withdrawn at any time without disturbing the others. If fitted with nuts, there is great difficulty in getting the tube out, generally necessitating the withdrawal of a whole row, while there is really no difficulty in tapping the holes, and no necessity for nuts. The stay tubes in the Navy have a *plus* thread at each end, the diameter of the front end being larger than that at the back; in this case, the thickness at bottom of thread is the same as that of the body of the tube.

Stay tubes are $\frac{1}{4}$ to $\frac{1}{2}$ inch thick in the body, and as the thread is $\frac{1}{16}$ inch deep, they are $\frac{3}{16}$ to $\frac{7}{16}$ inch thick at the bottom of thread. Some makers prefer to fix thick tubes, and space them farther apart. For 100 lbs.

working pressure and upwards, the stay tubes are $\frac{1}{4}$ to $\frac{3}{8}$ inch thick, each alternate tube should be a stay tube—that is, in a nest of, say, 64 tubes, there will be 16 stay tubes.

Stay tubes generally outlast two sets of the ordinary tubes.

Serve tubes, manufactured by John Brown & Co., Sheffield, have a series (6 to 8) of longitudinal ribs running the whole length of the tube inside and projecting inwards about $\frac{3}{4}$ inch, thus forming an additional absorbing surface for the heat. Besides the additional surface, this tube causes a better circulation of the hot gases within it. In small boilers especially is this tube a useful one, as, with a large grate area, a sufficient heating surface can be obtained.

Retarders.—By introducing into the inside of a plain tube a twisted strip of metal of the same breadth as the inside diameter, Howden found a considerable gain in efficiency, owing to the hot gases getting stirred and circulated in their passage through it.

Stays.—Flat surfaces have to be stiffened and tied together by bars called *stays*. When the surfaces are close together and the plates comparatively thin, so that the stays are short and numerous, they are screwed into both plates, and the ends either riveted over or fitted with lock nuts; such stays are usually called "screwed stays." As has been said, the thickness of plates and pitch of stays are interdependent. The size and number of the stays depend on the pressure they have to withstand. The stays in the steam space must be so spaced that a man can pass between them, and for this purpose they should never be nearer than 14 inches, centre to centre, and are usually 15 to 17 inches centres, which gives a clear space of 12 to 14 inches between them. These stays are seldom more than 3 inches effective diameter, and as the exact spacing of them depends on the form and size of the boiler, they are generally arranged to suit the particular case, and the diameter varied to give a section adequate to the load each has to bear. To admit of easy access, these stays are arranged in horizontal and vertical rows as nearly as possible.

The Admiralty allow a stress of 18,000 lbs. per square inch of effective area of stay when at the test pressure, which is usually 90 lbs. above the working pressure. If the stays are below $1\frac{1}{2}$ inches diameter, only 16,000 lbs.

The Board of Trade allow 9,000 lbs. per square inch at working pressure.*

Lloyd's Registry allow, on stays not exceeding $1\frac{1}{2}$ inches smallest diameter at working pressure, 8,000 lbs. on "screwed" stays and 9,000 lbs. on others; on stays above $1\frac{1}{2}$ inches smallest diameter, 9,000 lbs. on "screwed" stays and 10,000 lbs. on others.

British Corporation Rule is $D = \sqrt{\dfrac{S \times W}{C}} + \frac{1}{8}$.

S = the surface in square inches supported by the stay.
D = the effective diameter of stay.
W = the working pressure.
C = 8,000 for steel screwed stays; 8,900 for steel longitudinal stays.

* Tested wrought iron of 21·5 tons tensile may be used for screwed stays instead of steel, if preferred. Longitudinal stays may be of same quality as shell plates and stressed to one-sixth the ultimate strength at working pressure.

Bureau Veritas has the same rule, but C = 300 × tensile strength of metal in tons, which, for iron, is 22 tons, and for steel according to test.

The large stays are sometimes made with a *plus* thread; this necessitates "upsetting" the ends so that the body is of somewhat smaller diameter than at the bottom of thread of the ends. This is a somewhat expensive process, and is not so reliable as simply screwing a rolled bar with a *minus* thread at each end. The latter plan, especially since the making of steel boilers has become general, is now fast taking the place of the former; it has, too, the advantage of excess of section in the body where most corrosion takes place. These stays are secured to the plate with a nut and washer outside, and a nut inside to lock, whose length is two-thirds that of the outside, which is one diameter long. When weight is of consequence, steel bars may be swelled and have a *plus* thread by "upsetting" the ends, as the Board of Trade now permit this.

The screwed stays are usually from $1\frac{1}{4}$ to 2 inches diameter, with a standard thread 9 per inch. The most useful sizes are $1\frac{1}{4}$, $1\frac{1}{2}$, and $1\frac{3}{4}$ inch, suitable for $\frac{7}{16}$ to $\frac{5}{8}$ inch plates, and pressures from 60 to 220 lbs. per square inch. When screwed through a plate whose thickness is less than half the diameter of the stay in length there should always be a lock-nut with a thin washer, the nut being two-thirds the diameter of the stay in length. The practice of "nobbling," or riveting over the ends of these stays, is very objectionable when they pass through thin plates, as extreme pressure is very apt to cause the stay to draw completely through the plate, and this is especially so when the plate is ductile and soft like mild steel. Screwed stays had fine threads, which were in accordance with Whitworth's rule as to the number per inch—viz., $1\frac{1}{4}$ diameter, 11 threads; $1\frac{1}{2}$ diameter, 10 threads; $1\frac{3}{4}$ diameter, 9 threads; and 2 inches diameter, 8 threads. It is, however, more convenient to have only one number for all sizes, so now the Engineering Standards Committee has decided on 9 per inch for all stays from $1\frac{1}{4}$ to 2 inches; ordinary stays above 2 inches, 6 per inch, with nuts on each side of the plate.

When several stays are fitted with nuts and washers at their ends, the following rule holds good:—

$$\text{Pitch of stays in inches} = 11 \times \sqrt{\frac{(\text{Thickness of plates in sixteenths})^2}{\text{Working pressure}}}.$$

Example.—What pitch of stays is suitable for a plate $\frac{5}{8}$ inch thick for a working pressure of 160 lbs.?

$$\text{Pitch} = 11 \times \sqrt{\frac{10^2}{160}}, \text{ or } 8{\cdot}73 \text{ inches.}$$

*Board of Trade Rule.**—Working pressure $= \dfrac{C \times (T + 1)^2}{S - 6}$,

T being thickness of plate in sixteenths of an inch.
S the surface in square inches.
C being 100 for screw stays with nuts, 165 for longitudinal with nuts and washers two-thirds the thickness of plate.

* For the new Rules of the British Marine Engineering Design and Construction Committee for stays and flat surfaces, *v.* Appendix.

Lloyd's Rule :—Working pressure $= \frac{C \times T^2}{P^2}$.

P^2 being the mean of squares of pitches in rows and between rows, and C and T as before.

C, for screwed stays with nuts, 110 with plates under $\frac{7}{16}$ inch thick; over $\frac{7}{16}$ and under $\frac{9}{16}$ inch, 120; over $\frac{9}{16}$ inch, 135; longitudinal stays, with double nuts and outside riveted washers, two-fifths the pitch in diameter, and half the thickness of plate, 200.

British Corporation Rule :—Working pressure $= \frac{C \times T^2}{P^2 + p^2}$.

Here P is the greatest and p the least pitch.

C is 265 for screwed stays with nuts, and 370 for stays with double nuts and outside riveted washers, the latter being two-thirds the thickness of plate and one-third the pitch in diameter.

Bureau Veritas Rule is similar to that of the British Corporation, but the value of C changes with the tensile strength of the plate, and for screwed stays with nuts 415·3 for 27-ton steel, and 461·5 for 30-ton. For longitudinal stays with inside and outside nuts and washers, the outside washers being 0·4 of the pitch in diameter, and two-thirds the thickness of plate, C is 435·4 and 483·8.

Flat plates may be stiffened to allow of wider spacing of the stays than given by this rule, by fitting thick washers of large diameter to each stay, or by connecting the stays to the plates by means of angle-irons or T-bars; the latter plan possesses the advantage of distributing the strain over a large area, and that without a doubtful joint, as is the case with nuts. The old plan of riveting a doubling plate of common iron in wake of the large stays has almost disappeared.

Continental Practice differs from British in two respects in the design and construction of marine boilers. Almost invariably the tops of the combustion chambers are in a horizontal line in whatever position they are placed in a ship; it is true that when a yacht or other vessel having sail power, and requiring to use it whenever possible, as is the case with cruisers in the Pacific and other extensive oceans where coal is scarce and dear, has the boilers placed axially fore and aft, it is desirable that the chamber tops shall be sloped, that when the ship is listed over under canvas the chamber top on the weather side shall not be in danger of emersion and exposure to steam only with the liability of overheating. In other ships with boilers placed in this way there is, of course, some liability to list, and when rolling the chamber tops may get their ends bare. In the latter circumstances it is of no consequence, as the exposure is only temporary, and the emersion only for very short periods, but with a list more or less permanent there may be some risk of damage. It is, however, very slight, and British engineers design the boilers under the belief that it is the duty of those in charge to keep a sufficient amount of water in the boiler under any circumstances to submerge the parts exposed to heat. If boilers are made with the combustion chamber tops sloped away, as is common in German practice (*v.* Fig. 210), there is considerable sacrifice of total heating surface as well as tube surface, and an increase in the weight of water in the boiler. On the other hand, with boilers made in this way there may be at all times less water over the chamber tops at middle than would be prudent with horizontal top ones.

Continental boiler-makers also adopt chain riveting frequently for shell-plate joints instead of zigzag, and when employing the special arrangement of treble riveting, whereby alternate rivets are omitted in the outer rows, the inner butt strap extends only to the double rows of closer-pitched rivets and the outer strap only is taken by the whole six rows of rivets. In this way the edges of the inner strap can be caulked more severely, and the effective rivet area per pitch is 8·50 times the section of one rivet, as against 9·37 when both straps are taken by the rivets, and a considerable weight of butt strap is saved. In the old days of small plates, the latter consideration had much more weight than it has to-day, when the largest boilers have six longitudinal joints, whereas in those days a small single-ended boiler would have that, and the largest double-ended ones 12 to 16 joints.

When weight is of very great importance, the butt straps can be vandyked instead of being straight at the edges, and when this is so the caulking can be heavier, as the distance apart of the rivets parallel to the edge is less than the pitch of the outer rows. (*v.* figs. on. p. 649).

This vandyking with the outer and omission of cover with the inner straps is followed likewise when the double-riveted joint is adopted with alternate rivets in its outer rows omitted. In this case the effective rivet area is 4·75, against 5·63, as commonly done. A few boiler-makers will even employ two sizes of rivet in these special joints with three rows, so that those of wide pitch in the outer row are of larger diameter than those in the inner rows. For example, with 10 inches pitch and $1\frac{1}{2}$-inch rivet the plate left is 85 per cent. of the solid plate; the inner row of rivets may be $1\frac{1}{4}$ inches diameter with the plate 75 per cent., instead of 70 per cent. with $1\frac{1}{2}$ inches.

It is, however, in practice very inconvenient to work with two sizes of rivet, and it is only under certain very special circumstances that it is worth while departing from the common well-understood practice.

Water Spaces.—The spaces between the furnaces themselves, between the furnaces and shell, and between the combustion chambers, although sometimes diminished, should not be less than 5 inches; that between the backs of combustion chambers and shell should taper from 6 inches at the bottom to 9 and even 12 inches at the top, to allow of the free current upward of the steam generated on the surfaces. If the spaces are less than 6 inches, it is very difficult to hold up rivets, to clean the surfaces from scale, or to get a good circulation.

The space between the nests of tubes should not be less than 10 inches, and, when possible, should be 12 inches. This permits a man to go down to clean across, and ensures good circulation.

Man-holes.—The chief one in the shell should be oval, 16 inches by 12 inches; those in the ends, etc., may be 15 inches by 11 inches; and the smallest through which a boy can pass is 14 inches by 10 inches. Mud-holes are generally 9 inches by 6 inches, and peep-holes 6 inches by 4 inches. The hole cut in the shell plating should be surrounded by a doubling plate or angle bar ring, to compensate for the metal cut away, and the holes through thin plates should have such rings to stiffen the edges. A better plan is to flange the plate inwards so as to stiffen the edge and give a good face for the

TABLE LXXX.—PARTICULARS OF MODERN CYLINDRICAL BOILERS.—SCANTLINGS.

	X.	A.	B.	C.	D.	E.	F.	G.	H.	K.
Diameter, mean,	17′ 8″	16′ 9″	15′ 3″	16′ 4″	16′ 2″	16′ 3″	15′ 6″	16′ 0″	15′ 0″	15′ 9″
Length over end plates,	22′ 0″	20′ 0″	18′ 6″	17′ 6″	11′ 0″	10′ 9″	11′ 0″	11′ 6″	12′ 0″	21′ 0″
Working pressure, lbs.	195	190	230	180	220	190	215	180	180	215
Rules on which designed,	B.T.	B.T.	B.T.	B.T. B.V.	B.T. Ll.	Admlty.	B.T. Ll.	B.T.	B.T.	B.T.
Number of furnaces,	8	8	6	8	4	4	3	3	3	6
Diameter of furnaces, external.	49·5	$44\frac{3}{4}$	$46\frac{1}{4}$	$42\frac{1}{2}$	$45\frac{1}{2}$	$46\frac{3}{4}$	$49\frac{1}{4}$	52	49	45
Thickness of furnaces,	0·655	..	$\frac{23}{32}$	$\frac{18}{32}$	$\frac{22}{32}$	$\frac{19}{32}$	$\frac{23}{32}$	$\frac{21}{32}$	$\frac{20}{32}$	0·687
Number of combustion chambers,	8	4	3	4	4	2	3	3	3	6
Number of tubes,	1,064	968	644	840	438	578	396	342	463	812
Diameter of tubes,	$2\frac{1}{2}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$
Length of tubes,	8′ $0\frac{1}{4}$″	7′ 8″	7′ 4″	6′ 11″	7′ 8″	7′ $5\frac{1}{2}$″	7′ $2\frac{1}{4}$″	7′ 9″	7′ 6″	8′ 1″
Surface of tubes,	5,480	4,860	4,017	4,170	2,417	2,820	2,044	2,254	2,272	4,733
Surface of other parts,	1,120	1,010	660	861	544	467	462	516	465	969
Total heating surface,	6,600	5,870	4,670	5,031	2.961	3,287	2,506	2,770	2,737	5,702
Grate area,	169	180	121	160	90	94·2	..	72	67·8	131
Thickness of shell,	1·60	$1\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{21}{32}$	$1\frac{17}{32}$	$1\frac{21}{32}$	$1\frac{7}{16}$	$1\frac{11}{32}$	1·672
Diameter of shell rivets,	1·63	$1\frac{19}{32}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{21}{32}$	$1\frac{17}{32}$	$1\frac{21}{32}$	..	$1\frac{7}{16}$	..
Pitch of shell rivets,	10·5	$10\frac{1}{2}$″	9″	10″	10″	10″	10″	..	$9\frac{11}{16}$	..
Tensile test of shell plate,	..	30	36 to 40	26	31 to 34	28 to 31	29 to 32	29 to 32	29 to 32	..
Thickness of top end plates,	..	$1\frac{5}{32}$	$1\frac{1}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	..
,, front tube plates,	0·875	$\frac{7}{8}$	1	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	..
,, back plates,	0·812	..	..	..	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	..
,, combustion chambers,	0·655	..	$\frac{23}{32}$	$\frac{19}{32}$	$\frac{5}{8}$	..	..	$\frac{11}{16}$	$\frac{5}{8}$	..
Weight, as finished, tons	..	..	..	78·5	57·0	52·0	53·0	..	..	107
Weight per 100 square feet T.H.S.,	..	..	..	1·560	1·926	1·582	2·115	..	..	1·877

door joints; this is now very general. The doors are usually placed inside the boilers, and held to their faces by studs screwed into them, which pass through strong cross bars or "dogs," held by square nuts; the main door in the shell is, however, sometimes fitted externally, and connected to a flanged ring with bolts, in the same way as a steam chest door on the engines. The doubling ring is, in this latter case, formed of a very thick angle-iron, whose *deep web* is flanged to fit to the boiler, and whose flange forms the face for the door.

The doubling plate at a shell man-hole is now as a rule flanged inwards, and the edge of the flange machined for the door to fit on.

The weight of boiler is approximately $= \dfrac{D^2 \times L \times \sqrt{p}}{F}$ tons.

For single-ended boilers with a chamber to each furnace, F is 740.
For double-ended boilers with two furnaces to each chamber, F is 810.
For double-ended boilers with a chamber to each furnace, F is 780.

The total heating surface is approximately $= D^2 \times L \times K$ square feet.

For single-ended boilers, K is 0·97 to 1·15.
For double-ended boilers, K is 1·0 to 1·10.

The higher value of K in each case is with small tubes well packed; the lower value is for the boilers of ordinary merchant steamers with at least 3-inch tubes. In the above both L and D are in feet.

CHAPTER XXV.

BOILER MOUNTINGS AND FITTINGS.

Smoke-Box.—This appendage to the boiler is for the purpose of receiving the products of combustion as they emerge from the tubes, and conducting them through the "uptakes" into the funnel. In the old box form of boiler, it was built inside, and formed an integral part of the boiler; but, with the modern cylindrical boiler, it is a separate structure, secured to the boiler front by studs. It is constructed of iron or steel plates and angles of "ship" quality, and made "smoke tight" only; it should, however, be caulked if necessary, to prevent air passing to the inside. The advent of cheap mild steel permits of flanging instead of angles. In front of the tubes are a number of doors hung on hinges, and so arranged that every tube may be swept or removed in case of necessity. The doors should be rigid, and of such a size as to be easily handled, and when the nests of tubes are so large as to cause the door to be too ponderous if made in one, two doors may be fitted to close on a portable stanchion. The doors are sometimes arranged to open on a horizontal and sometimes on a vertical axis; the latter is preferable when possible, as then they are more easily handled.

The bottom of the smoke-box should be at least 12 inches broad, measured in direction of the length of the boiler, and, when possible, as much as 15 inches. If too narrow, it is soon filled with soot and ashes, so as to cover the ends of, and render useless, the bottom rows of tubes; the baffle plates on the doors are also soon burned away. The bottom plate should be at least 2 inches below the bottom row of tubes, and the side plate the same distance from the side tubes, so that the tubes may be drawn clear of the 2-inch angle-iron rim around the doorways. The front of the smoke-box is sloped outwards, so as to be about twice the breadth of the bottom from the boiler front, above the level of the top row of tubes. Above this, the smoke-box contracts towards the funnel base, and its configuration must depend on the position of this, and on the consideration that the section transverse to the flow of gases must have an area at least equal to the area through the tubes. The part between the smoke-box and funnel is called the "uptake," or "take-up"; it should have easy bends, and lead as directly as possible to the funnel, and be without recesses and obstacles where eddies are formed and the draught may be baffled.

The bottom and sides of the smoke-box and sides of the uptake should be of $\frac{1}{4}$-inch plates for large boilers, $\frac{3}{16}$ for smaller ones, and where weight is of more consideration than endurance $\frac{1}{8}$ is sufficient. The smoke-box doors should be of the same thickness, and have baffle plates $\frac{1}{10}$ inch thick on the inside, and air or screen plates of the same thickness on the outside; these screen plates prevent radiation of the heat to the stoke-holes, and, for the same purpose, the sides of the smoke-box and uptake should be fitted in the same way. When steel is used, there may be a reduction of 20 per cent. in these thicknesses, and, in naval ships, the outer casings may be $\frac{1}{16}$ inch only. Steel sheets are now cheaper than iron, and will stand flanging, bending, and working in a way that is impossible with iron; consequently iron is now not used for such purposes as these.

To protect the boiler front, which has only steam on its inner surface, the uptake should have a back commencing from just above the level of the

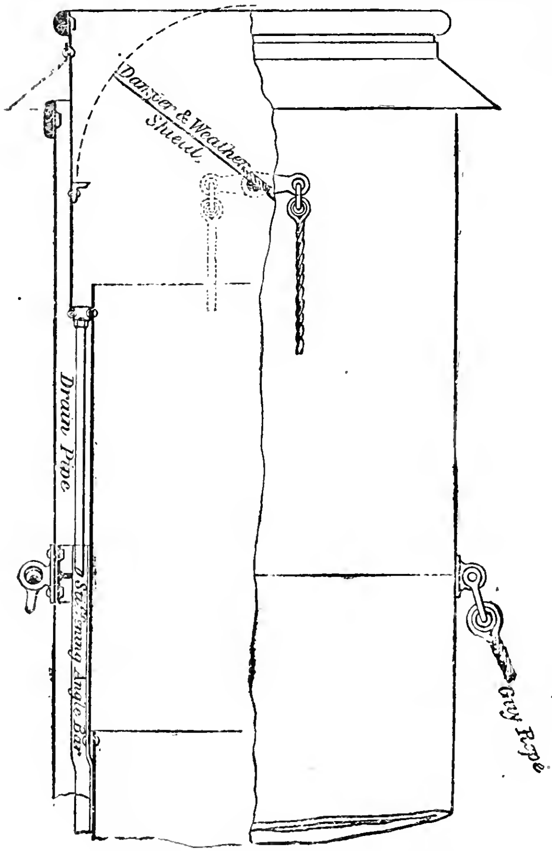


Fig. 253.—Naval and Yacht Funnel Top.

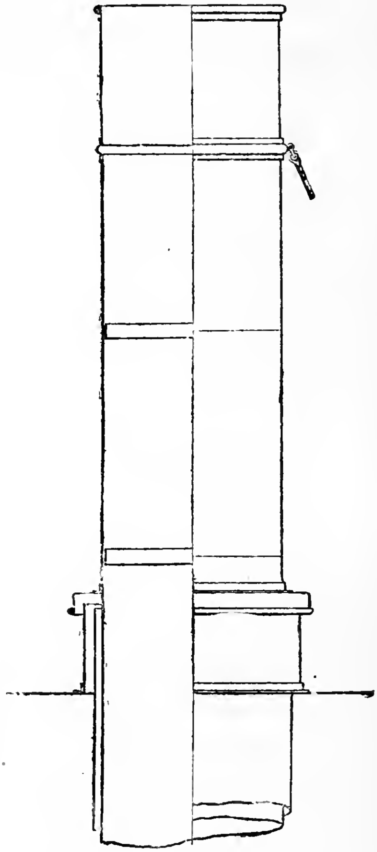


Fig. 254.—Ordinary Funnel, with Hood, etc.

top row of tubes. When this cannot be done, a good and well-fitting baffle plate should be fixed to the boiler front.

A casing is also fitted round the funnel from its base to the level of the deck casing, to prevent radiation. A corresponding casing is fitted round the funnel, above the level of the deck coamings, to a convenient height, and over this is fitted a hood secured to the funnel, so as to prevent water passing down, while allowing the hot air to come out. This hood is called by various names, as "cravat," "bonnet," etc. (fig. 254). It is now often at the top of the funnel, so the latter is completely protected from the weather (fig. 253).

Where there are several boilers discharging smoke to one funnel, each smoke-box should have a separate uptake, so that the smoke from one does not enter the box of another; and, when there are no good ashpit doors, there should be a damper in each of these uptakes, so as to regulate the draught and get a uniform evaporation from all the boilers, and, in case of necessity, to isolate a particular boiler.

TABLE LXXXI.—Pitch, etc., of Riveting for Funnels, Casings, etc. (Admiralty Work).

Description.	Thickness of Plate.	Diameter of Rivet.	Pitch of Rivet.
	Inches.	Inches.	Inches.
Uptakes,	$\frac{3}{16}$	$\frac{3}{8}$	2
	$\frac{1}{4}$	$\frac{1}{2}$	2
	$\frac{5}{16}$	$\frac{5}{8}$	$2\frac{1}{2}$
	$\frac{3}{8}$	$\frac{5}{8}$	$2\frac{1}{2}$
Funnels,	$\frac{1}{4}$	$\frac{1}{2}$	3
	$\frac{3}{16}$	$\frac{3}{8}$	2
	$\frac{1}{8}$	$\frac{3}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Funnel casing,	$\frac{3}{16}$	$\frac{3}{8}$	3
	$\frac{1}{8}$	$\frac{3}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Deck casings,	$\frac{3}{16}$	$\frac{3}{8}$	2
	$\frac{1}{8}$	$\frac{3}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Cowls and trunks (exposed to weather),	$\frac{1}{8}$	$\frac{3}{8}$	2
	$\frac{1}{16}$	$\frac{1}{4}$	2
Screens and ventilating trunks,	$\frac{1}{8}$	$\frac{3}{8}$	4
	$\frac{1}{16}$	$\frac{1}{4}$	3
Other work,	$\frac{3}{16}$	$\frac{3}{8}$	2 to 3
	$\frac{1}{8}$	$\frac{3}{8}$	2 to 3
	$\frac{1}{16}$	$\frac{1}{4}$	2

Funnel.—This is usually of circular section, but sometimes, to minimise the transverse size of the boiler-hatch, it is made of oval section. The funnels of men-of-war are often made of oval section for the same reason, but, instead of the section being an ellipse, as is sometimes the case in the mercantile marine, it is like that of an oval boiler (fig. 204). The best height to look well is 5 to 7 diameters above the taffrail, the latter when there are high bridges or boats in wake of the funnel. For the same reason, the ring for the shrouds should be $\frac{9}{10}$ the diameter from the top. Funnels are made of ship quality plate, lap-jointed, or butt-jointed with single straps inside; the latter costs more, but when so made is more durable. Another method much in fashion at one time, and which presents a good appearance, is to make the longitudinal joints with inside butt-straps, and the circumferential with a band of iron of a flattened **U** section (fig. 254).

The funnel plates should, for strength, be thicker at the base than at the top, but the top plates wear out faster than those at the bottom. The following may be taken as the approximate thickness of these plates:—

Top plates	= 0·1 inch	+ 0·025	for each foot of diameter.
Middle ,,	= 0·125 ,,	+ 0·026	,, ,,
Bottom ,,	= 0·15 ,,	+ 0·027	,, ,,

If the funnel is stiffened with angle or **T** bars (fig. 253), it may be made of somewhat thinner plates. The funnels of naval ships are, of course, made as light as possible, and the plates composing them are seldom more than $\frac{3}{20}$ inch thick, and of steel.

Furnace Fronts and Doors.—Although often made of cast iron, they are better when of wrought iron to withstand the rough use to which they are exposed. It is from this cause that all the improved doors which have been tried have been finally rejected; and because of this rough usage all attempts at refinement in the fittings, etc., meet with want of success on board ship. The smaller the door the better, as, when open, an excess of air passes into the furnace and lowers its efficiency; on the other hand, it must be large enough to stoke, work, and clean the fires properly. A long grate requires a larger door than does a short one. Furnaces of large diameter—that is, above 42 inches—should have a pair of doors to be used alternately. The amount of opening when stoking or cleaning fires is thereby reduced, and the sides of the grate are better attended to.

The doors should be so arranged as to remain open in a seaway when required; this may be effected by making a projection and corresponding recess in the hinge. A star damper should be fitted to the fire door, by which, when open, a supply of air is admitted to the fires. The baffle plate inside the door should have a number of small perforations in this case, to finely divide and to distribute the extra supply of air.

Fire-bars.—The length of grate should never exceed twice the diameter of the furnace, as the fires cannot be properly worked when this is the case. To get the highest efficiency of grate, it should not be more than $1\frac{1}{2}$ times the diameter. The slope of the grate should be 1 inch to the foot, which may be increased, in furnaces over 42 inches diameter, to $1\frac{1}{2}$ inch with advantage. If the grate is over 5 feet long, there is generally some difficulty in properly stoking the back end, and it is only a good fireman who can properly work the fire on a long grate. The increased slope materially helps to overcome this difficulty, and at the same time the fire is better supplied with air at the back, and not choked by the products of combustion from the front.

The bridge or brick barrier at the end of the grate should be built to such a height that the area of passage over it is not less than $\frac{1}{8}$ nor more than $\frac{1}{6}$ the grate area; and, when possible, the distance from the top of the bridge to the top of the furnace should be sufficient for a man to pass into the combustion chamber.

When South Wales or other similar coal is to be burnt on the grate, there should be no space between the bars and the side of the furnace; when the furnaces are corrugated, these side bars should be made to fit into the corrugations.

The fire-bars are usually in two lengths; but the grate is more efficient when they are in one, as the bearer is avoided, which baffles the free flow of air to the fire above it, and prevents the fireman from "pricking" effectively. Cast-iron fire-bars to burn bituminous coal may be 5 feet 6 inches long, but if the coal contains much sulphur they are safer in two lengths. The Admiralty do not allow the bars to be longer than 27 inches, even when made of wrought iron, as they generally burn Welsh coal, and may have to use, on foreign stations, coal containing sulphur.

Fire-bars are made from $\frac{7}{8}$ to $1\frac{1}{4}$ inch broad on the face; the former is better when the bars are not very long, and when of wrought iron may with advantage be even $\frac{5}{8}$ inch. In the mercantile marine $1\frac{1}{4}$ inch is the usual breadth when the bars are long.

The depth of the bar at the middle depends on the length, and should be—

$$= 0{\cdot}6\sqrt{\text{length}}, \text{ when of cast iron,}$$

and

$$= 0{\cdot}5\sqrt{\text{length}}, \text{ when of wrought iron.}$$

The thickness at the bottom should be one-third the breadth at the face, and should taper to two-thirds beneath the flange.

For burning bituminous coal there should be a space of $\frac{1}{2}$ inch at least between the bars, and, when it cakes quickly, there may be as much as $\frac{3}{4}$ inch; but, if Welsh coal or American anthracite is to be burnt, there should never be more than $\frac{1}{2}$ inch spaces, and with narrow bars the space may with advantage be less.

Martin's Patent Bars consist of wrought-iron bars of square section placed with the angles upward, and so arranged that the bars may be slightly turned so as to clean the fires.

Henderson's Patent Door and Bars.—This is one of the most successful of the improved grates. The bars are of cast-iron and ordinary section, hung in a frame, which can be easily moved by means of levers, so as to slightly move alternate bars longitudinally; this movement, while carrying the fuel gradually to the back of the furnace, breaks up the clinker, and obviates the necessity of cleaning the fire. The door of the furnace is hinged horizontally at the bottom, and so arranged as to drop into a recess in the dead plate left for it, which recess is filled by a back piece attached to the door, and turns down as the door opens; if the fire needs cleaning, the door is turned still farther, so as to leave the gap in the dead plate, and form a slope below it to shoot the clinker and cinders into the ashpit instead of raking them on to the stokehole floor. The furnace front, too, is carefully designed, so as to pass a current of air completely about it between the front and back, thus serving the double purpose of keeping the front comparatively cool, and of heating the air before entering over the fire. This arrangement of grate when properly worked permits of a rapid and complete combustion of coal equivalent to a moderate forced draught, as may be seen by referring to Tables lxxxvii. and lxxxviii. where grates are marked "H."

To Burn Crude Petroleum Residuals and other similar liquids, a special burner is necessary, and on its efficiency depends the economy of using such fuels.

The functions of such an apparatus are to direct the fluid in a steady stream to the place where ignition commences; to pulverise it, or "spray" the stream so as to permit of quick and perfect ignition; to project the flame in the required direction, and to keep it constantly going with the least possible expenditure of steam, air, or anything else which costs money.

Mr. James Holden, formerly the locomotive-superintendent engineer of the Great Eastern Railway, after experimenting for a considerable time with the burners used on the Caspian Sea and in America, devised an instrument of his own, which has been found to give the greatest satisfaction, both on locomotives and on shipboard, by performing its duties efficiently and without fail over considerable periods. Steam is employed to drive the liquid through the nozzle (fig. 201), but, before doing so, it has induced a flow of air in with it from the central tube, which air mingles with the pulverised liquid and helps to burn it completely, as well as to furnish the oxygen required for ignition. The ring around the nozzle is charged with steam, and this forms a heater for the air current, which flows past it to the nozzle. There are, however, other burners which do not require steam to work them (*vide* Sir F. Flannery's paper in *Trans. I. Naval Architects*, vol. xliv.). Fig. 201*a* is a good example of a simple burner worked by compressed air, and found to be very efficient on shipboard. In this case the nozzle can be cleared from obstruction by opening the central valve and admitting steam or air.

Stop-valve.—The main stop-valve should be of sufficient size to pass out all the steam the boiler is capable of making *with little resistance*, but no more than all. The chief loss of pressure at the cylinders is often due to the stop-valves being only partially open; on the other hand, when the cylinders are large and the strokes of the piston comparatively slow, priming may be effectively checked by partially closing the stop-valve, so that there is no sudden withdrawal of steam, *in fact, the valve should be set so as not to pass out more steam than the boiler is making.*

The area of opening, sufficient to pass all the steam a boiler can make can be obtained from the following rule:—

$$\text{Area of orifice} = \frac{\text{total heating surface} \times \text{R}}{8 \times \text{P}}.$$

R is the rate of evaporation per square foot of heating surface per hour, P is the pressure absolute.

The area through a pipe to carry away the steam is two to three times the net area, depending on the length (Tables xxxii. and cxi.).

The diameter of the stop-valve is often settled from considerations of the size of the main steam pipe at the engines. Let D be the diameter of the main steam pipe, and n the number of boilers (at least two), then

$$\left.\begin{array}{r}\text{Diameter of branch pipe}\\ \text{to each boiler}\end{array}\right\} = \text{D}\sqrt{\frac{4}{3n}}, \text{ or } \frac{2\text{D}}{\sqrt{3n}}, \text{ or } 1{\cdot}156 \times \frac{\text{D}}{\sqrt{n}}.$$

The area of pipe section to suit a boiler may be found by the following rule :—

$$\left.\begin{array}{l}(0{\cdot}25 \text{ square inch per square foot of grate} + 0{\cdot}01 \text{ square} \\ \quad \text{inch per square foot of total heating surface})\end{array}\right\} \times \sqrt{\frac{100}{\text{pressure}}}.$$

Example.—To find the diameter of steam pipe from a boiler whose grate area is 50 square feet, and the total heating surface is 1,500 square feet; pressure, 80 lbs.

$$\text{Area of section} = \{(0{\cdot}25 \times 50) + (0{\cdot}01 \times 1{,}500)\} \times \sqrt{\frac{100}{80}}$$
$$= 30{\cdot}7 \text{ square inches.}$$

Therefore the diameter should be $6\frac{1}{4}$ inches.

Example.—To find the diameter of the main steam pipe of a locomotive boiler whose grate area is 16 square feet, the total heating surface 1,200 square feet, and the pressure 150 lbs.

$$\text{Area of section} = \{(0{\cdot}25 \times 16) + (0{\cdot}01 \times 1{,}200)\} \times \sqrt{\frac{100}{150}}$$
$$= 13{\cdot}09 \text{ square inches.}$$

Therefore the diameter should be $4\frac{1}{8}$ inches.

The diameter of the stop-valve should be such that the clear area past it is not less than given by the above rules.

$$\text{The diameter of spindle} = \frac{\text{diameter of valve}}{50} \times \sqrt{\text{pressure}} + \tfrac{1}{8} \text{ inch.}$$

The valve and seat are of a bronze which should be both hard and strong at the temperature of the steam passing it (Chapter xxx.). The valve should have the boiler pressure always on the side opposite the spindle, so that it helps to open it. The spindle should have a square thread, and, when possible, the screwed part should be outside, so that in opening or shutting the valve the spindle does not turn round. As the full pressure is on the valve when shut, the bridge, etc., should be strong enough to withstand it. The seat, when fitted with wings for the guide to the spindle, should be carefully secured and the wings curved, so that, when expanding with the heat, the seat is not distorted. These seats, when fitted into cast iron or steel are very apt to get loose and leak from the permanent set of the metal, induced by the resistance of the cast iron to expansion of the bronze. They should, therefore, be secured by bolts or studs to the casting, and a joint made which will keep tight in spite of expansion, etc., much in the same way as the pipe joints are done.

The Admiralty require stop-valves and all other boiler mountings to be made of bronze or steel; for modern high pressures, and especially when the steam is superheated, the material must be such as to retain strength and elasticity with such high temperatures. The immadium bronze made by Parsons M. Bronze Company is found very suitable, as at 550° F. it has an elastic limit of 8 to 9 tons.

The Admiralty likewise require all stop-valves to be self-acting—that is, they shall close on the pressure in the boiler being decreased below that in the main steam pipe. This is with the object of localising the danger and automatically disconnecting the boiler in case of accident to it from shot, etc. Such valves have their spindles turned down at their outer ends, so as to pass through a hole in the screwed shank, fitted to the cross-piece, as shown in fig. 255, and with cross-handles to permit of the valve being turned or closed as required.

Safety Valve.—As its name implies, this valve is for the purpose of providing a safe and self-acting means of relieving the boiler from excessive pressure. A good safety valve should be (1) large enough in diameter, and have sufficient lift to allow the steam to escape as fast as it is generated, when the pressure is 5 per cent. above that to which the valve is loaded; (2) it should be made so that it closes again as soon as the pressure has dropped very slightly below the load; (3) it should be free to open and shut, so that it may always act efficiently and promptly; (4) it should be enclosed so that it cannot be tampered with or accidentally interfered with by pieces of coal,

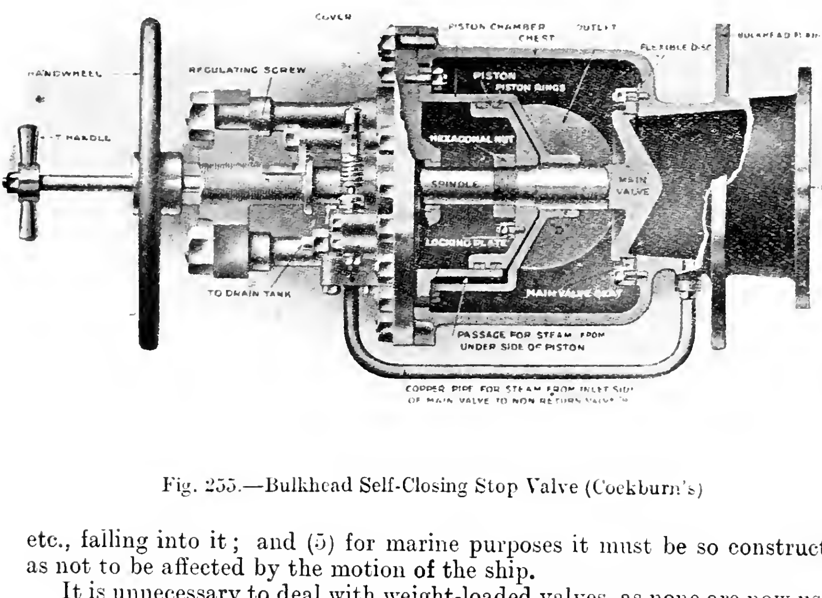


Fig. 255.—Bulkhead Self-Closing Stop Valve (Cockburn's)

etc., falling into it; and (5) for marine purposes it must be so constructed as not to be affected by the motion of the ship.

It is unnecessary to deal with weight-loaded valves, as none are now used, the fifth condition being satisfied by means of steel springs for the load (fig. 256). When weights were used, the amount of lift given to the valve by the steam pressure was very small, and, since the lifting of the valve compresses the spring and increases the load, the spring-loaded valve opens somewhat less. This being so, area of opening can only be obtained by increasing the diameter of the valve. Many ingenious methods of increasing the lift have been tried, but all those involving the use of special mechanism have given place to those which do without it; the most successful and best known of the latter is Richardson's Patent (fig. 257), generally called *Adams'*, after the name of the manufacturer, who purchased, improved, and worked the patent in this country, and Cockburn's Valve (fig. 258), designed and made by Messrs. Cockburn, of Glasgow. These consist essentially of an ordinary mushroom valve with a secondary outer rim of U section which overlaps the rim of the seat, so that there is a second contracted orifice at the outer edge of this rim. As soon as the valve opens, the steam fills the

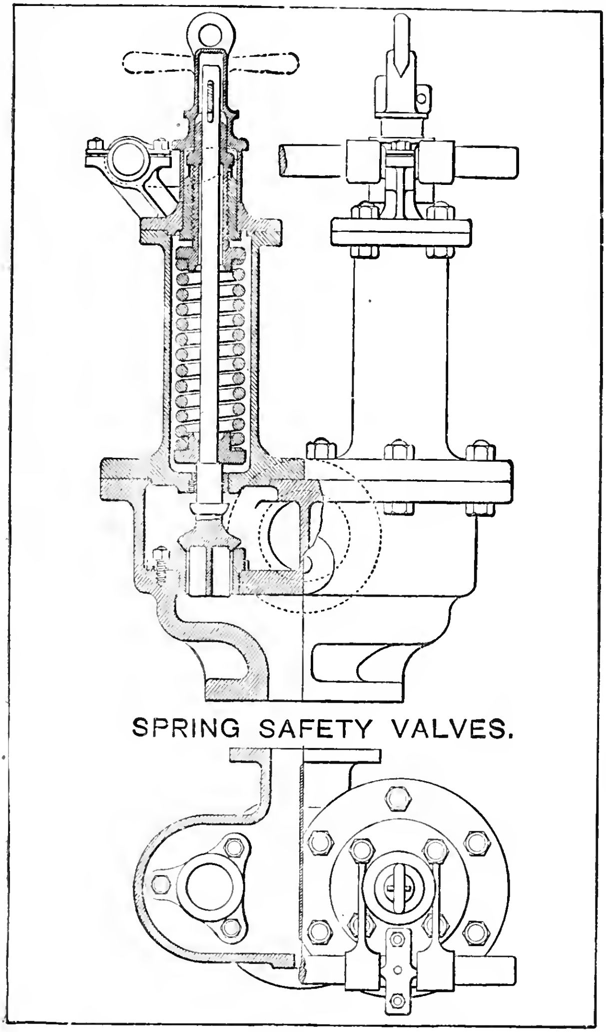


Fig. 256.

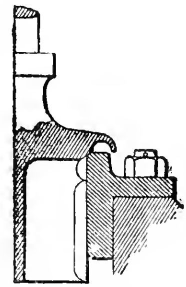


Fig. 257.—Adams' Valve.

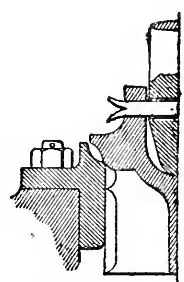


Fig. 258.—Cockburn's Valve (Navy Type).

outer rim, and the valve is then virtually of larger area; the load on it is so suddenly increased that the valve lifts wide open immediately, and will continue to vibrate with the spring until the pressure falls so as to be insufficient to open the valve when the latter touches the seat in one of its vibrations and remains closed. The second condition is best fulfilled when the valve can be made to dance over its seat from the vibration of the spring. The third condition can only be fulfilled by making every movable part a very easy and, in most cases, *a very slack fit*, and having springs whose resistance to compression is *uniform on every side*; this is a quality not always found in the springs supplied commercially, and adjusting a spring when badly made is expensive and troublesome; to avoid the evils due to this cause, Messrs. Cockburn have devised a joint for the spindle which shall adjust itself to suit the irregularity of pressure when the spring is compressed as

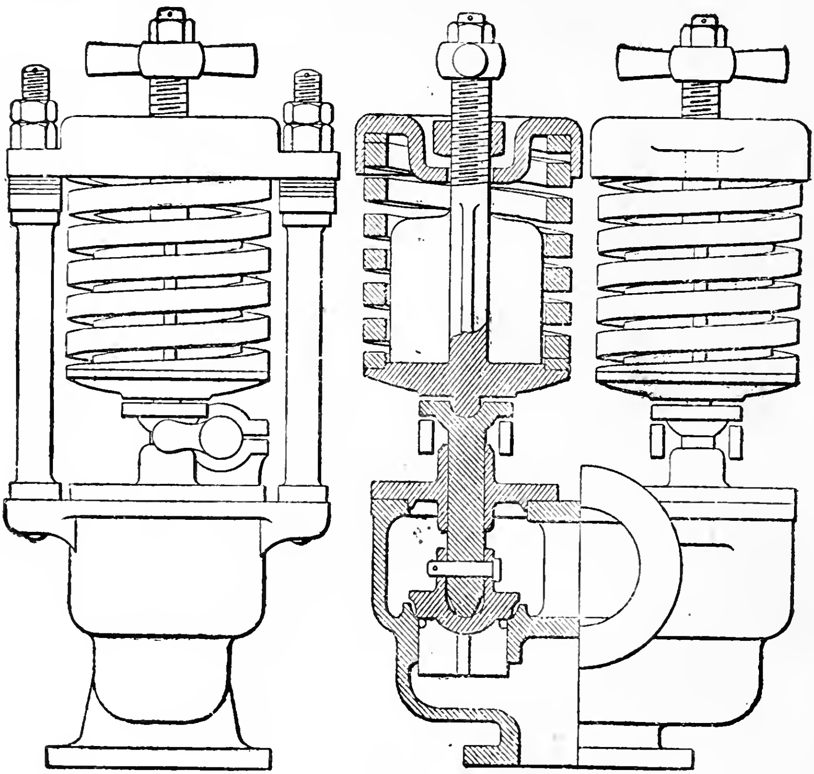


Fig 259.

shown in fig. 259, which is the kind of safety-valve fitted on naval ships where the head room is very limited. Fig. 260 is the design of safety-valve now largely used in naval ships, and made by Cockburn & Co.

To prevent the valve from being injured by accident or design, it should be enclosed in a case, and the Board of Trade require that such cases shall be locked up, and the key kept by the captain of the ship.

The Size of Safety Valve.—This must suit the *volume* of steam which can be generated by the boiler in a given time, and that depends on the weight of fuel it can consume, on its efficiency, and on the working pressure. In similar boilers—that is, boilers made on the same general design, and worked with the same draught and pressure—this volume of steam varies with the grate area. The original rule laid down by the Board of Trade for the area of safety valve is based on this, and it was found to work satisfactorily; since then, however, new rules have been laid down. Strictly speaking, any rule for the safety valve should fix the amount of *circumference* rather than area, and considerable allowance should be made for the load pressure, as for the same weight of steam the volume varies inversely as the pressure.

The following are the rules for the size of valve:—

(1) To satisfy the *Board of Trade*, the valves must not be in any case less than 2 inches in diameter, and for each boiler with natural draught, the area of valve or valves combined must not be less than given by this rule.

$$^{*}\text{Area of safety valve in square inches} = \frac{37\cdot5}{P} \times \text{grate area (square feet)},$$

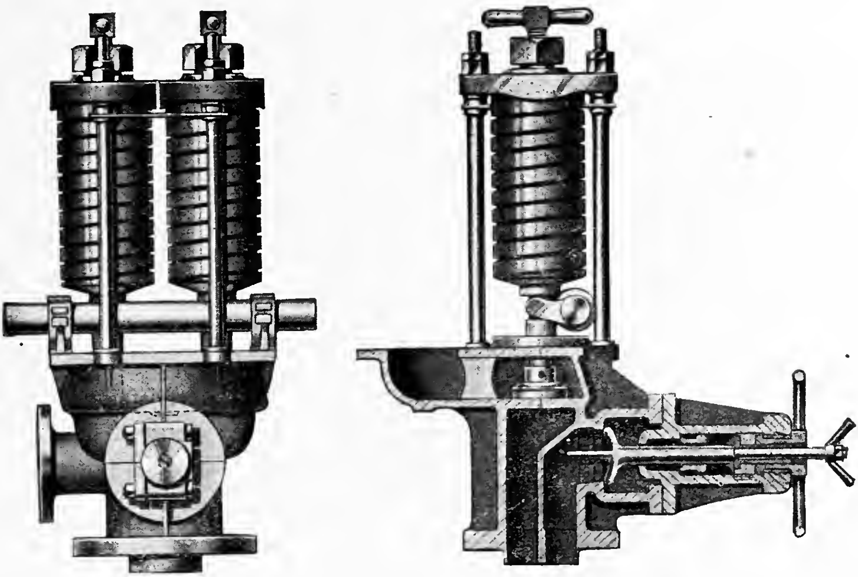


Fig. 260.—Combined Stop and Safety Valves, Admiralty Type (Cockburn's).

P being the *absolute* boiler pressure. For forced draught the area found by the above rule must be multiplied by $\frac{C}{20}$, where C is the estimated coal consumption in lbs. per square foot of grate per hour.

(2) *Lloyd's* retain the old Board of Trade Rule that the valve area must be at the rate of ½ square inch for each square foot of grate, but allow special valves of any size, so long as they be satisfactory when tested.

(3) The *French Government Rule* is based on the amount of heating surface contained in a boiler, and this, perhaps, is the truest gauge of a boiler's capability, as it bears a constant relation to the amount of coal consumed; allowance is also made for the steam pressure. The diameter is in inches,

* To suit all conditions of draught and fuel it is better to take a hypothetical grate area of equal to one twenty-eighth of the total heating surface.

the heating surface in square feet, and the pressure in lbs. per square inch.

$$\text{Diameter of valve (if only one)} = 1{\cdot}23\sqrt{\frac{\text{total heating surface}}{\text{pressure} + 9}}.$$

(4) The *German Government Rule* also makes allowance for the steam pressure, and is as follows:—

To have a *clear* area of valve or valves, after deducting for the wings or other obstacles, at the rate of so many *square millimetres* for each square metre of total heating surface, in accordance with the following table:—

Working pressure in atmospheres	5	6	7	8	9	10
Number of square millimetres per metre of surface	131	112	98	86	79	72
Working pressure in atmospheres	11	12	13	14	15	16
Number of square millimetres per metre of surface	66	60	56	54	52	51

(5) An improved rule, which is simple and easily used, is—area of each of two valves = (0·05 square inch per square foot of grate + 0·005 square inch per square foot of total heating surface) × $\sqrt{\frac{100}{\text{pressure}}}$.

Example.—To find, by the various rules, the size of a pair of safety valves for a boiler whose grate area is 45 square feet, heating surface 1500 square feet, and working pressure 180 lbs.

(1) By Board of Trade.

$$\text{Area} = \frac{45}{2} \times \frac{37{\cdot}5}{195} = 4{\cdot}33 \text{ square inches.}$$

Therefore *the diameter is* $2\frac{3}{8}$ inches of each valve.

(2) By French Government rule.

$$\textit{Diameter} = 1{\cdot}23\sqrt{\frac{1500}{180 + 9}} = 3{\cdot}47 \text{ inches.}$$

(3) By German Government rule, for one valve.

$$\text{Clear area} = 60 \times 1500 \div 10{\cdot}76 = 8365 \text{ square millimetres.}$$

$$\text{Net area} = \frac{8365}{645} = 13 \text{ square inches.}$$

Add to this 2 square inches for obstruction of wings,

gross area = 15 square inches,

and diameter for one valve is 4·4 inches, for each of two valves 3·1 inches.

(4) By the improved rule.

$$\text{Area} = (0{\cdot}05 \times 45 + 0{\cdot}005 \times 1500) \times \sqrt{\frac{100}{180}} = 7{\cdot}3 \text{ square inches.}$$

Therefore *diameter is* 3·1 inches.

The mitre on a safety valve-seat should not be more than $\frac{1}{16}$ inch broad, except for very large valves, and the bearing area in any case need not exceed that necessary for a pressure of 1200 lbs. per square inch on it when there is no steam pressure on the valve; hence

$$\text{Breadth of mitre} = \text{diameter of valve} \times \frac{\text{working pressure}}{4800}.$$

The Board of Trade rule for the size of steel for the spring is

$$d = \sqrt{\frac{S \times D}{C}}.$$

S is the total load on the valve; D the diameter of coil, measured from centre to centre of wire, in inches; d is the diameter of round wire, and the side of square section wire; C is 8,000 when the coil is made of round section steel, and 11,000 when of square section (*vide* Appendix O).

Internal Pipes should be fitted from the stop-valves to the highest part of the boiler, and be made with holes or slits, whose collective area is equal to twice the area of section of the pipe.

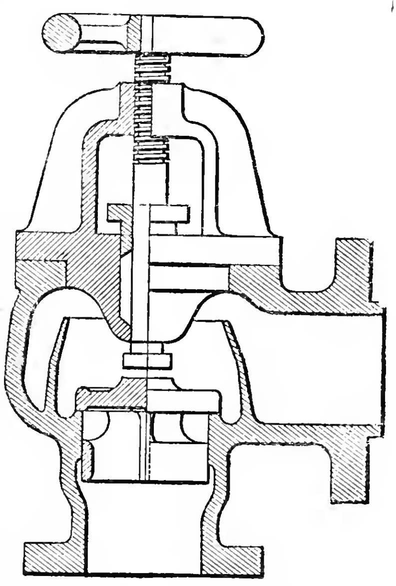


Fig. 261.—Improved Feed Check Valve.

The chief object of this pipe is to collect the steam gently from every part of the boiler, so as to avoid setting up a strong current in one particular direction, and thereby induce priming. These pipes are usually made of brass, but some engineers prefer copper, and others made them of cast iron to avoid the risk of galvanic action and reduce the cost. Now that salt water is never used for feed water, these pipes may be of steel galvanised. By fitting an internal pipe, the stop-valve can be placed in a position convenient for examination and working, and it should always be situated so as to be easy of access at all times. Arrangements should also be made for opening and shutting it without going into a position of danger or difficulty, and this can always be effected by lengthening the spindles or fitting chain gear. The Admiralty and some passenger-ship owners insist on having gear fitted so that the stop-valves can be shut from on deck as well as in the stokeholes.

In the mercantile marine, the stop- and safety-valve boxes are almost invariably made of cast iron or steel; the valves, seats, and spindles being of bronze. The Admiralty require all boiler mountings to be made of bronze or cast steel, and do not allow cast iron to be used.

Feed-valves.—Each boiler should be fitted with a self-acting, non-return valve, through which the main feed-water is pumped. It should also have a screw spindle, which may be used to regulate the lift, or to shut it down when water is not required. There should also be a similar valve through which the auxiliary pump can discharge water to the boiler.

The valve is generally of mushroom form, and made similar to the ordinary stop-valve, except that it is detached from the spindle. It is made wholly of bronze, and should be very strong, as at times the pressure on it may be excessive. An inner casing should surround the valve (fig. 261), and be formed or pierced in such a way that the flow is fairly even round the periphery, otherwise the valve tends to cant and wear its seat unevenly, with leakage as a consequence. Locomotive engineers sink the feed check valve seat well below the discharge orifice for this purpose.

There should be 2 square inches of clear area through the valve and pipe for every 100 lbs. of water evaporated per minute; or, put in a more convenient form—

$$\text{Area of pipe section} = \text{pounds of water evaporated per hour} \div 2.500,$$

$$\text{or} \quad \text{diameter feed pipe} = \sqrt{\text{pounds evaporated per hour}} \div 45,$$

or area through main feed-valve in square inches

$$= \text{total heating surface in square feet} \div 250;$$

and, area through donkey feed-valve in square inches

$$= \text{total heating surface in square feet} \div 300.$$

When feeding with an independent pump the pipes may be 30 per cent. smaller, as the velocity of flow can be varied.

As the feed-valves cannot always be placed on that part of the boiler best suited to receive the feed-water, and also in order to distribute that water so as to avoid its affecting the boiler plates, an internal pipe should be always fitted. To prevent the necessity of blowing the boiler down in case of accident to the feed-valves, it is a very common practice to fit these valves high up on the boiler, even in many cases above the water-level. This plan also has the advantage of providing a means of warming the feed-water, than which nothing is more essential for the preservation of the boiler; the heating is effected by the passage of the water through a long internal pipe of brass or copper, which leads it to where there is a down current of water, so that the comparatively cold feed-water may not interfere with the circulation. To permit of the check valves being examined and ground in without emptying the boilers, a stop valve is fitted, as shown in fig. 262, and proved to be very useful, especially in ships making long voyages.

Some engineers prefer to inject the feed-water in the form of spray, either above or a little way beneath the surface of the water in the boiler; this avoids all chance of injury to the boiler plates, as any gaseous matter mechanically mixed with the feed-water is at once given up and mixes with the steam.

Great care should be taken in any case that the internal feed-pipes "run full"—that is, that they are never at any time filled or partly filled with steam, but always with water.

The *dynamic* effect of the steam on the feed-water, when mixed inside the pipe, is very startling; every stroke of the feed-pump delivering cool water produces a concussion by the sudden condensation of the steam, so that, in a very short time, both external and *internal* pipes are damaged seriously.

To avoid this, the internal pipe should, when discharging above the water-level, be *turned upward* at the end, so as to always remain filled with water; and when turned downward to discharge under water, the end should be well below the lowest working level.

An additional means of safety is sometimes afforded by fitting inside the boiler a clack valve, so arranged as to close over the end of the internal pipe or on the spigot of the ordinary check valve; when this is provided, the latter can be examined when steam is up. A cock also used to be fitted sometimes close to the check valve, so that the supply can be regulated by it, instead of by interfering with the lift of the check valve.

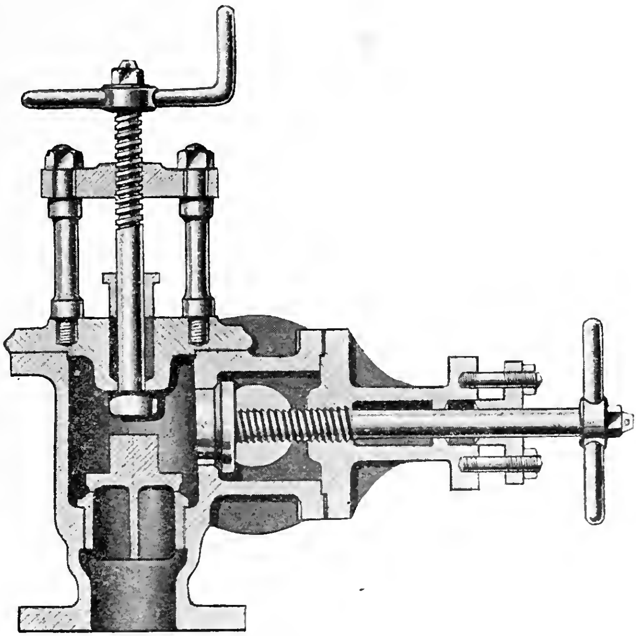


Fig. 262.—Feed Check Valve with Shut-off Valve Added.

Automatic Feed Valves are an advantage to all water-tube boilers, and necessary for their good working when steaming under forced draught, inasmuch as the ratio of the water they contain to that evaporated per hour is small; consequent on this, there might be considerable difficulty in maintaining the water at the right level by hand adjustment, as is effected with tank boilers. It is also necessary for their good working and the production of the full supply of dry steam that the level be as low as possible consistent with safety. Fig. 262*a* shows the method for accomplishing this desirable end devised by Mr. A. G. Mumford, and the apparatus supplied by him, and used largely in the British and foreign Navies. The means involved are simple, effective, and free from liability to derangement with ordinary care and attention; moreover, if out of order, there is no difficulty in restoring it. The check valve on the boiler has attached to it a piston, which is a leaky fit in a cylinder below the valve box, and the opening or shutting of the valve depends on the difference in pressure below from that above the piston. So long as the water which leaks past the piston is free to run away, the

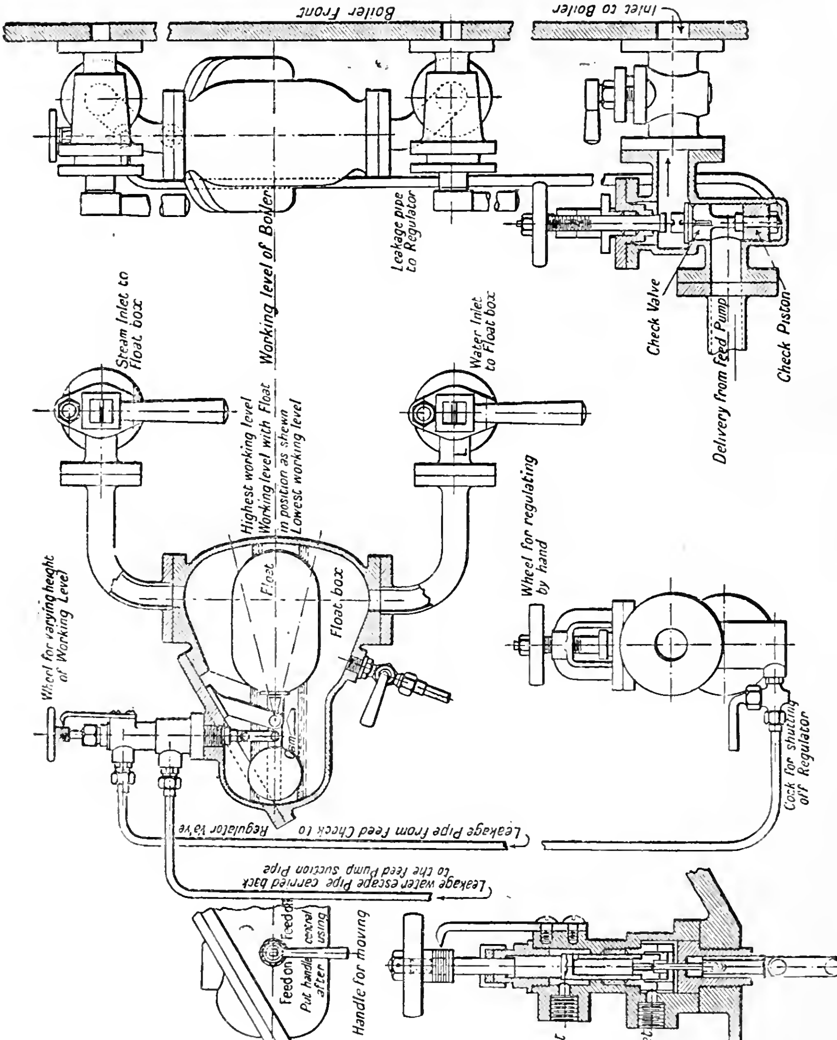


Fig. 262a.

pressure of the feed-water is unable to open the check valve. If the flow from below the piston is checked, there is soon equilibrium, and the feed-water forces open the check valve and supplies the boiler; as soon as the flow is resumed, the piston will be pressed down, and so close the check valve and stop the feed. The water thus leaking away is caused to pass through a small valve, which is operated by the rising and falling of a float in a chamber connected with the boiler, so that on the water level falling below normal the leak is stopped, and the check valve permitted to open and remain so until the level is at or slightly above normal when the leak is free and the check valve closed.

There are other methods of attaining the same end, such as that of Sir John Thornycroft, where the admission valve is operated direct by a float inside the boiler, so that the supply is shut off as soon as the water level rises above normal; and the ingenious arrangement of Mr. Yarrow, whereby the feed-pump steam cylinder became flooded as soon as the level was too high in consequence of its supply valve being placed at the proper level for good working; the water was exhausted from the feed donkey to the feed tank, and as soon as the water level fell and steam was again admitted the donkey increased to its regulation speed, and delivered its normal supply of feed-water.

Blow-off Cock.—A cock used to be fitted at or near the bottom of the boiler, to answer the double purpose of admitting sea-water before getting up steam and to *blow off* brine and some of the water when required. This cock should be a very strong one, as it is liable to rough usage, and, being out of sight and not easily got at, it is very apt to be neglected. For this reason, as well as because a large cock is difficult to open and shut, engineers now prefer a valve to a cock. If, however, a cock is fitted, it should be arranged so that its handle or spanner cannot be removed when it is open. Sea-water is now seldom or never used for filling boilers working at high pressures, so that a connection with the sea is not necessary; when fitted, the clear area through it was usually 1 square inch + (0·2 square inch for each ton of water in the boiler), now it need not exceed 1·5 inches in diameter.

As it is a very reprehensible practice quickly to blow off a marine boiler when at its normal working temperature, it is now never done; a cock is fitted, however, to the bottom of the boiler, so that, when the pressure of the steam is down, the water may be pumped overboard if required.

Scum Cock.—A cock, having a clear area through it of one-third that of the blow-off cock, should be fitted to the boiler, near to the level of the water, and to it is connected a perforated pipe, inside the boiler, not lower than the lowest working level. The object of this pipe is to collect all *scum* and floating impurities from the water and discharge them overboard. Considerable quantities of grease and greasy matter were pumped with the feed-water into the boiler, and had to be got rid of occasionally. A particular kind of grease was sometimes formed in the condensers of engines whose cylinders were lubricated with a certain class of oils which are not pure hydrocarbons; portions of it were pumped with the feed-water into the boiler in the form of small pellets, which, being of superior specific gravity to pure water, sank to the bottom, and used to remain there until the density of water increased sufficiently to cause it to rise and come in contact with the hot surface. It is, therefore, better to filter the feed-water under all circumstances than to trust to a scum cock or to a blow-off cock, and to use little or no internal lubricant; if a lubricant is used at all, care should be taken that it is specially suited for the purpose and a pure hydrocarbon (*vide* Chapter xxxi.).

The scum cock was used as a means of reducing the quantity of water in the boiler before adding a fresh supply from the sea; but, if the surface was clear of dirt, this was better done with the bottom blow-off, especially if it was possible to check evaporation for a few minutes before blowing off.

Water Gauge.—It is of the first importance that those in charge of a boiler shall know with certainty the position of the water-level within the boiler. The ordinary gauge for this purpose consists essentially of a glass tube, whose ends communicate freely with the inside of the boiler, and so situated on the boiler that the plane of the water surface bisects the tube transversely when at its normal working level. It is, however, found necessary in practice to use considerable discretion in the choice of position of this gauge. Since a difference of one-tenth of a pound pressure corresponds to 2·7 inches of water, it is quite possible so to place the gauge as to give very false readings. The upper end of the gauge should not communicate with the boiler near to any exit for steam, for the rush of steam past the orifice can easily make a reduction in pressure of one-tenth of a pound in the gauge pipe. The lower end should also be clear of any part from which steam is evolved, as steam bubbles might flow into the pipe, and tend to raise the water-level in the glass. It should also be well away from the currents due to circulation or other causes, and as near to "dead" water as possible.

It is usual, especially with large boilers, to fit the gauge cocks and *test cocks* to a brass casting connected by pipes to the bottom and top of the boiler; this is called a "stand pipe," and is a necessity when the gauge is on the front of the ordinary boiler. The test cocks are placed on the stand pipe at the lowest and highest working levels, for the purpose of checking the glass gauge, and for use when the latter is broken or out of order. In these days of high pressures, however, they are practically of no use, and might well be done away with and their price go towards an additional glass gauge.

The length of the gauge glass visible should be at the rate of $1\frac{1}{4}$ inch for each foot of diameter of the boiler; the external diameter of the tube is $\frac{5}{8}$ inch for small boilers and $\frac{7}{8}$ inch for large ones; the glass is usually about $\frac{1}{8}$ inch thick. The Admiralty use $\frac{5}{8}$-inch glasses for all sizes of boiler.

The gauge is so placed that the water is just disappearing, or, as it is generally said to be just "in sight," when the level is from 2 to 4 inches above the top of the combustion chamber; the allowance, however, should be 0·3 inch for each foot of diameter of boiler.

The pipes connecting the stand pipe to the boiler should be from 1 inch to $1\frac{1}{2}$ inch diameter, and of strong copper, so as to be fitted direct to the boiler. The Board of Trade and some engineers insist on having a cock on the boiler at the top and bottom; but this, like many another intended extra safeguard, is itself a real source of danger, for the cocks are apt to be shut by mistake or carelessness, and thus cause the gauge to show a false level. That this is no mere fanciful danger has been proved on more than one occasion.

All large boilers, and those in ships which are often under sail, should have two water gauges placed as far apart as possible in an athwartship vertical plane.

The gauge and test cocks should always be fitted with a small plug in line with the bore, which, on being removed, allows a wire to be introduced to clean it of deposit and scale.

Some engineers have the stand pipes and connecting pipes so arranged that there is a continuous flow from top to bottom *only via the glass tube*, the stand pipe being divided in the middle, or being so formed that the middle is only a connecting bar for top and bottom to prevent the glass tube being drawn out. This is very simple, and safer than the ordinary method.

Steam Gauge.—The steam gauge on Bourdon's principle is now nearly universal and so well known as to need no description. Schœffer's original gauge, although less liable to derangement than Bourdon's, is not so accurate. and does not find so much favour. The boiler gauge should have a dial so marked that it may register pressures to at least 25 per cent. higher than the working pressure of the boiler. These gauges should be carefully tested when new, and at frequent intervals after being at work, as it is often found that they require some slight adjustment. For this purpose all long-voyage ships should have a standard gauge and test apparatus.

Sentinel Valve.—The Admiralty used to require each boiler to have a

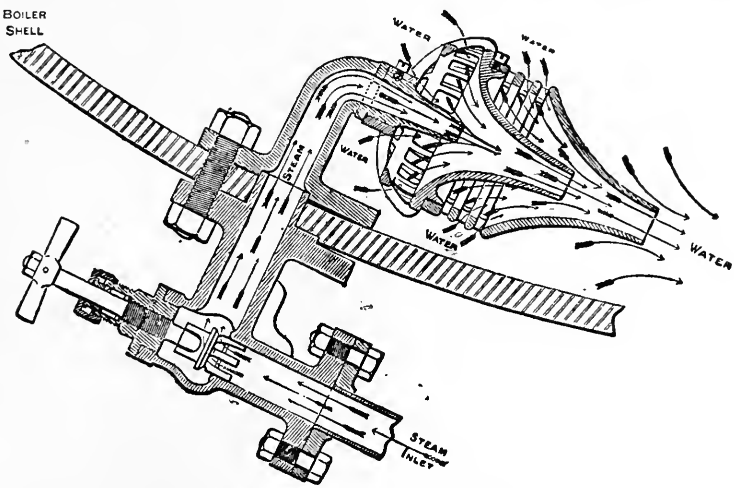


Fig. 263.—Weir's Hydrokineter.

small valve loaded with a weight to a few pounds per square inch above the working pressure, so that in case of the safety valves sticking fast and the gauge being false, an alarm may be given when there is an excess of pressure. Such valves are generally about $\frac{3}{4}$ inch in diameter, but sometimes as small as $\frac{3}{8}$ inch. An arrangement of a small safety valve attached to a whistle has been introduced, but there should be no necessity for such refinements, and it is doubtful if, in time of need, they would be heeded.

Air Valve.—The old box boilers had a small non-return valve to admit air when the boiler was cooling down to prevent the boiler collapsing from excess external pressure To-day such a valve or cock is desirable to admit air before taking off manhole doors, and avoid accident to the operator.

Weir's Hydrokineter.—This instrument (fig. 263) is for the purpose of warming the water in the bottom of the boiler when getting up steam. It consists of a series of nozzles, one within the other, each having a grating-body in rear, through which the water passes on its road to the nozzle, when a current is set up by a jet of steam issuing from the centre one. The steam is obtained from the auxiliary boiler, which has been used to supply the winches. Without this instrument the bottom of a large boiler remains cold, even after the steam is raised; with it the temperature of the water at the bottom differs very little from that at the top; steam can in this way be safely raised in a shorter time than usual, and at no extra cost, and the

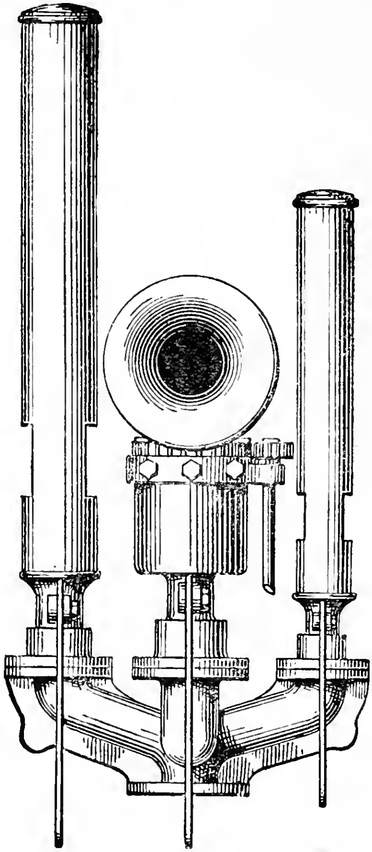


Fig. 264.—Combined Syren and Double Organ Whistle.

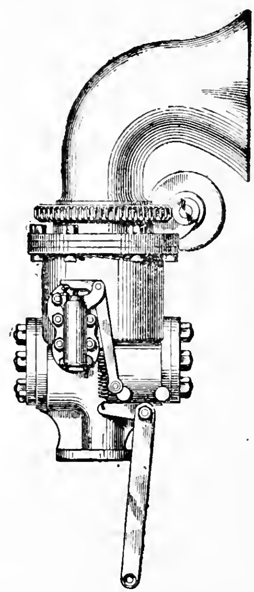


Fig. 264*a*.—Steven & Struthers' Admiralty Syren.

endurance of the boiler is very considerably increased. There are many other ways of promoting the circulation when steam is up, but none do this so efficiently during the time of raising steam as the *hydrokineter*.

Steam Whistles are of two kinds, known as the bell-whistle and organ-tube whistle; the latter has now superseded the former, on account of its simplicity of construction and superior tone. An improved form has a division in the tube, or two separate tubes, so as to emit two distinct notes which may be in harmony or discord, and when sounded together are heard a long distance.

It is important that the whistle shall sound as soon as the steam is turned on; to insure this happening, great care must be taken to keep the whistle-pipe free of water, which is no very easy matter. It may, however, be effected in two ways: first, by leading the pipe from the boiler into the funnel, and keeping it inside as far as the level of the whistle; second, by taking steam for the steering engine from the top of the whistle-pipe, thereby ensuring a constant flow of steam and no accumulation of water.

Separator.—This, although not a boiler fitting, is intimately connected with them; it is almost unknown in the mercantile marine, although it might be used sometimes with advantage there in order to free the steam from water mechanically mixed with it. All men-of-war were formerly fitted with separators, and, from the tendency to prime on the part of their boilers when working at full speed, and the danger to the engines when working at a high velocity of piston from water getting into the cylinders, they were necessary. Now, only ships with certain water-tube boilers have separators.

The old separator consisted of a vertical cylindrical chamber, having a division-plate extending from the top to about half-way down, and so placed that the steam in going through the separator must pass under this diaphragm: the object is to separate out the water mechanically mixed with the steam, by dashing it against the diaphragm, and precipitating it to the bottom of the separator, whence it is blown to the hot-well or sea, whichever is convenient. Most modern separators are constructed on the centrifugal principle, so that the water by its great density follows one course, while the steam follows another.

Boiler Clothing.—The boiler shell should be well covered with a coating of non-conducting material, to prevent loss by radiation from its surface, which may amount in some cases to more than 10 per cent. The material used should be a non-conductor of heat as well as incombustible and inorganic. The following materials* are those in general use for boiler clothing:—

* Some experiments on the efficiency of different heat insulators are described by Mr. S. H. Davies in a recent issue of the *Journal of the Society of Chemical Industry*. The following table shows the heat transmitted through 1 foot thickness of the different materials for each 1° F. of temperature between the hot and cold surfaces:—

Air-Dried Material.	B.T.U. per Square Foot per Hour.	Weight of Medium per Cubic Foot.	Cost per Cubic Foot.
		lbs.	d.
1. Slag wool (light), ...	0·054	8·6	8·3
2. Hair felt, ...	0·058	7·9	20·6
3. Light magnesia, ...	0·062	10·1	26·0
4. Granulated cork, ...	0·069	6·1	9·8
5. Slag wool (heavy), ...	0·070	32·9	31·8
6. Kieselguhr, ...	0·073	15·0	9·6
7. Flaky charcoal, ...	0·082	14·5	7·8
8. Pumice (½-inch mean diameter), . .	0·095	25·0	24·1
9. Sawdust (spruce), ...	0·096	13·1	0·5
10. Asbestos fibre, ...	0·136	14·5	137·0
11. Sawdust (very moist), ...	0·298	...	...

Mr. Davies notes that powdered pumice is much inferior to pumice in small lumps.

Silicate Cotton or *Slag Wool*, manufactured from slag and having the appearance of cotton, is eminently fitted for boiler clothing. It is a good non-conductor, incombustible, and imperishable from chemical action; it is, however, very brittle, and for this reason will not withstand mechanical action; and, therefore, if loosely packed and subject to vibration, it soon becomes dust, which is most offensive if it gets into the engine bearings.

Asbestos Fibre has very much the same nature as silicate cotton, it is not so efficient, is more costly, but is more durable. A paste made with this material and magnesia is very efficient.

Magnesia, either alone as cement, or mixed with asbestos fibre into a paste, is a very excellent covering, and, moreover, is very light in weight, so that an extra thickness of it weighs no more than does an ordinary coat of other compositions of a similar nature.

Cements of various kinds are used, their efficiency depending generally on the amount of vegetable fibre contained in them. These have not so high an efficiency as the foregoing.

Fossil Meal, Kieselguhr, or infusorial earth, is a composition consisting of large quantities of minute shells, and forms a most efficient covering; besides being inexpensive, it is also incombustible and durable, and comparatively light.

Papier Maché is employed for this purpose, and is a fairly good material, but it is not altogether incombustible and is apt to rot, besides being heavy.

The last three materials must be put on when the boiler is hot, and be carefully done; this does not militate in their favour, as it is an objectionable thing to have steam on the boilers when the ship is being finished.

The silicate cotton and asbestos fibre may be covered with sheet iron. The cements are generally tarred over so as to be waterproof, and the parts exposed to wear covered with sheet iron or lead.

No wood should be used for clothing when it is possible to avoid it, as it so soon rots, and is at all times liable to take fire.

Cameron's Patent Lagging is an ingenious arrangement of wrought-iron framework, *strapped* to the boiler, so as to form a series of segments, which may be filled with any non-conducting substance, and is covered in with squares of corrugated sheet iron, secured in such a way as to be easily and quickly removed for examination or repair. This is the most perfect plan, as it avoids all piercing of the boiler shell with studs and the use of combustible materials, and while capable of being well secured, it is such as to be wholly removed and replaced in a very short time.

A much thicker coating of lagging is necessary with the high temperatures now common, and no boiler, pipe, or cylinder can be considered properly clothed if the temperature at the surface is more than 30° F. above that of the atmosphere surrounding it. The bottoms of boilers should be as well covered as the tops; in fact, no heat should be allowed to escape to the air from any part of the boiler that can be kept in. Even the uptakes and smoke boxes may, with advantage, be coated.

CHAPTER XXVI.

FITTING IN OF MACHINERY, STARTING AND REVERSING OF ENGINES, ETC.

Fitting Machinery into the Ship.—As soon as the building of the ship is sufficiently advanced to allow the engineers to commence their work, a line should be stretched in the place intended for the axis of the shafting, and from it reference lines must be scored on the bulkheads, sternposts, and other convenient places for future guidance, and to enable the shipbuilders to set the engine seating and tunnel pedestals with some degree of accuracy.

For this purpose piano wire answers best, as it can be drawn exceedingly tight without breaking, and the amount of "sag" is very slight, and does not vary. When wire 20 L.S.G. is used, and the tension on it is as much as it will bear with safety (about 200 lbs. is sufficient), the "sag" will not exceed 0·15 inch per 100 feet. At the sternpost and bulkhead holes, cross pieces of wood should be fixed and the centre transferred to them; with these centres circles should be "scribed" in and marked with a centre punch, when they serve as guides in boring out for the stern tube. It is sometimes found advantageous to verify the centre line markings by the system called "sighting"; this is done by placing battens horizontally at convenient places, whose upper edges touch the centre line of shafting, and, when viewed, should all coincide if they are exactly in line. A similar system of battens should then be placed vertically with the same result, if the line is straight. This plan, however, is a somewhat tedious one, and by no means reliable in most instances owing to the deceptive nature of the light in the hold of a ship. In long ships a surveyor's telescope may be used to check the centres.

Boring the Sternpost. — A boring bar with tool head, etc., is fixed accurately in position by means of the reference circles before-mentioned, and the sternpost, which has been roughly bored to within about 5 per cent. of the finished size before being fixed in place, is bored out to the exact size required; the bulkhead, with its liner, is also bored out, and also any other part into which the stern tube is required to fit accurately. The finishing cut through the sternpost should be commenced *from the inside*, as the wearing away of the cutting edge of the tool causes the hole to be slightly taper, and this allows the tube to be made a very tight fit in it.

Engine Seatings.—The superstructure raised on the ship's frames to carry the engines is called by various names, such as *engine bed*, *engine seatings*, *engine foundation*, *engine bearers*, etc., and is one requiring some skill to design and care to manufacture properly. As the success and efficient working of an engine very materially depend on this structure, too much care cannot be devoted to its construction.* The weight of a marine engine is considerable and concentrated on a comparatively small area; in a seaway

* Recent investigations of the source of trouble experienced in some naval ships in America with shafting, etc., have confirmed this view.

the inertia forces cause very severe stresses on the seating and on the bolts connecting the engine to it; and, in some cases, the strain of the engine itself when at work is borne largely by the bed on which it rests, owing to the want of rigidity in its own bedplate.

The ship cannot always be viewed as a rigid structure, for elasticity is observable in all ships, especially when unloaded, and is very marked in those built of considerable length for shallow water navigation. For this reason, the engine seating must be so designed as to add materially to the stiffness of the ship's structure, and be of sufficient strength to distribute any stresses caused by the weight of the engine over the main framework of the ship. To this end, the vertical portions of the engine seating should be worked in or, better still, be in one with the floor plates and keelsons, and the longitudinals should extend beyond the immediate vicinity of the engine bed, so as to distribute the load over a longer portion of the ship, and not localise it on a few frames. The longitudinal strength thus added to the ship's bottom should not cease abruptly at the bulkhead, as is commonly the practice, but be continued beyond it and decreased gradually. The effect of stopping the engine seating at the aft bulkhead of the engine-room, is to cause a sudden change of flexure in the ship's bottom at that point, when the ship is steaming in a heavy sea; this change of flexure will produce abnormal stresses on the shafting, especially on the after part of the crank-shaft; the after bearing of crank-shafts shows this by its tendency to heat, and, in extreme cases, the shaft is broken at that crank-arm, or at its junction with the crank-arm. When the crank-shaft was connected to the thrust shaft by drivers, the working of the ship was manifested by the squeaking sound emitted by them when not oiled.

The scantlings of the engine seating should never be less than those of the ship's floor plates to which it is fixed, and when the seating is high, they should be in excess of the ship's scantlings. The angle-irons should be carefully fitted, and the riveting more than usually good; the rivet holes should be fair and well filled with the rivet; and if the holes are not fair, simply drifting them out to allow the rivet to pass through is not sufficient; these remarks particularly apply to the connection of the top plate with the verticals. The top plate should be at least 50 per cent. thicker than the vertical plates, and well bedded in place. Fifty per cent. of the rivet holes should be drilled through and through, and the remainder rimered fair and true to ensure a thorough trustworthy foundation for a modern engine.

It was the practice with some engineers at one time to hold the engines down to the ship by a few large bolts which were connected to strong cross-bars under the reverse frames of the ship; this is, however, not a good practice, as the load is localised to an unnecessary degree, and such bolts are very apt to corrode from the action of bilge-water, and, being unseen, to break eventually without being discovered.

The engine seatings are peculiarly liable to decay from the action of bilge-water and its gases acting on the warm metal; to prevent this they should be carefully protected by cement where practicable, and well painted where cement cannot be got to stick; cement wash is, however, better than paint, and if mixed hot, and brushed on, will form a very efficient covering.

Thrust Block Seating.—This also, from its importance and the nature and the magnitude of the strains on it, must be carefully constructed. There should be three vertical plates, extending over, at least. four frames in small ships, and six to eight frames in large ones ; the centre plate should be above, and strongly secured to the keelson by angle-irons, its thickness should be 50 per cent. more than that of the floor plates ; the side plates should be 25 per cent. thicker than the floor plates, and connected to the reverse frames by strong angle-irons ; all three verticals should be caused, if possible, to abut on the engine seating and tied to it. The top plate should be of the same thickness as that of the engine seating, and when possible in line with it. Stop plates should be riveted to the top plate to serve for the thrust block to abut on. All the rivet holes in this top plate should be drilled quite fair with the holes in the angle-irons and connections, and care should be taken that all the holes in the thrust seating should be rimered fair and all rivets quite fill the holes. In ships of very large power, the base of the thrust block seating should extend over more frames, and the vertical plates should be worked intercostal with the floors, so as to form a direct tie to the ship's bottom plating.

Pedestals for Tunnel Shafting.—The plummer blocks for the tunnel shafts rest on the tunnel bottom when the shafting is not high, but when the distance is too great for this, pedestals are built of plates, whose thickness is about the same as that of the floors, connected by angle-iron, so as to form a stiff column ; the top plate should be 50 per cent. thicker.

Boiler Seatings or Bearers.—The boiler, with its fittings and mountings, together with the water it contains, requires a very strong support and efficient means of keeping it in place when the ship is rolling or pitching. When the boilers are placed athwartships (that is, their axes are athwartship), the bearers act as beams to distribute their weight over a large number of frames, and may be made of H section, so that the lower flange is riveted to the reverse frames, and the top flange carries the chocks, which are wedge-shaped, and shaped to fit under the boilers, and form saddles for them to sit in. Single-ended boilers should have two such saddles, whose breadth of face where the boiler rests should not be less than 9 inches for very large boilers, and 6 inches for small ones. Double-ended boilers of moderate length and size may have three such saddles, but long or large double-ended boilers should have four sets. The chocks are sometimes made of angle-iron and plates, but they are better made of cast iron or cast steel ; when of the latter materials, the patterns can be tried in place after the boilers are in position, and made in such a way that the chocks, when cast, will fit with sufficient accuracy as to require no packings ; when made of wrought iron, much expense is incurred in trying to make them fit, and in the end packing is often necessitated.

If the boilers are placed " fore and aft "—that is, with their axes longitudinally—the bearers are generally laid on the top of individual floors, thus localising the weight on a few frames only. To avoid the straining action proving dangerous, the longitudinals of the ship in wake of the boilers should be increased, and extra connections made between them and those frames carrying the bearers. Sometimes the boilers when in this position have been carried by longitudinal bearers inclined so as to be in planes passing through the axis of the boiler. Such bearers distribute the load over a considerable number of frames, but do not so well support the boiler, and moreover prevent access to the boiler bottom for examination and repair.

Boilers are now, except in shallow ships, placed well above the bottom plating and bilges, so that there is room for cleaning, examining, and painting, and permits easy access to the double bottom. There is also much less liability to corrosion from bilge water.

Fitting Machinery on Board the Ship.—The stern tube and the screw shaft are fitted into place, and all sea-cocks and valves fixed to the skin of the ship before it is launched. After it is in the water, the tunnel shafting is placed in position piece by piece, each one being set so that its coupling comes fair and true with that of the preceding one; the shaft bearings are raised on temporary packings until the whole of the shafting is in place and coupled up; when this is done, the shafting should be turned around so that the bearings, if not placed in *exact* position at first, may adjust themselves; the bearings should now have the proper packings fitted to them, and when bolted down, the coupling bolts should be withdrawn, and each shaft tried around to see that there is no want of correspondence at the couplings. This may seem a somewhat tedious process, but it is a very safe one, and one which prevents all possibility of shafting being fitted out of line; the engineers of the ship should occasionally withdraw the bolts, and prove the shafting true, especially after the ship has had cause for straining. Of course, the accuracy of this method depends on the care with which the couplings have been turned; but, as modern appliances are capable of turning a shaft flange quite true with very ordinary care, there is little cause for fear on this ground.

The engine bed-plate, or foundation plate, with the crank-shaft in place, is now lowered on board, and placed on temporary packings of iron; it is, by means of jacks, brought to its exact position, and proved by the shaft couplings as before. Permanent packings of *cast iron* are now carefully fitted between the temporary ones and the latter withdrawn, and their places filled with hardwood (teak, greenheart, elm, mahogany, etc.) packings formed of pairs of wedge pieces driven from opposite sides.

The Holding-down Bolts should be carefully fitted so as to distribute the load over the bed, and means provided to prevent the nuts from slacking back; the Admiralty require check nuts to be fitted to all holding-down bolts, but it is better to prevent movement of the nut by slightly riveting the bolt end, than to trust to check nuts.

Staying Engines.—Vertical engines, especially those of long stroke, are supposed to require some means of support to prevent undue straining of the columns when the ship is rolling heavily. Such engines, when supported by *vertical wrought-iron* columns, do occasionally show signs of such a need; but it is better to bear this contingency in mind when designing the columns, and make them sufficiently strong and rigid to withstand such strains, than to trust to getting support from the elastic hull of the ship. In case of a collision, such supports, if rigid, might be a positive source of danger, as the force of the blow delivered in their neighbourhood might seriously damage the engines, if not destroy them altogether.

Such support as is sufficient can be provided by splaying out the front columns, as shown in fig. 32, and making them of strong cast iron or cast steel.

Ramming Chocks and Stays.—The Admiralty require all vertical engines to be fitted with non-rigid stays from the cylinders to some part of the ship in such a way as to keep the columns, etc., from being damaged or strained when ramming the enemy. Boilers also must have stays and chocks to keep them from being jerked out of place from ramming, or other inertia effect. As a matter of fact, ramming is no longer looked on as a possible nautical form of attack, but ships do collide accidentally, and sometimes take the ground suddenly.

If W is the weight in tons of a boiler, its water, fittings, etc., and K is the full speed of the ship in knots. Then—

Area of section of steel to resist fore and aft movement should not be less than $\frac{W \times K}{96}$ square inches.

Boiler Seats.—The boilers require to be carefully fitted in their seats, and secured there so that they cannot be displaced by the rolling or colliding of the ship. The boiler should require no loose packings when the chocks are of cast iron; but when the saddles are of wrought iron, and do not fit the boiler exactly, it is better to interpose iron packings at intervals, and fill in the spaces between with hardwood wedge pieces. To prevent movement longitudinally, "toe" plates should be riveted to the frames or other convenient part of the ship's structure, and these should be stiffened by angle-irons. The "toe" plates should stand about 6 to 9 inches above the bottom of the boiler. and be clear of the man-holes and mountings. The boiler is held sometimes in its seating by straps surrounding it, and secured to the bearers when of small size; but the general practice is to secure it by tie-bars from its upper part to the side of the ship, or by struts formed of plates and angles from the stringers, bearers, etc. Each particular case requires special treatment, and it is impossible to lay down any rule, beyond that of providing for every contingency to which a ship is liable. Tie-bars from lugs riveted to the boiler sides to the bearers is a good and perhaps the best way to secure them, and is the Admiralty practice.

At one time the Admiralty practice was to lay the boiler in a bed of mastic cement, spread on a cradle formed to suit the boiler bottom. This was very necessary for the box form of boiler, especially in wooden ships. This cement also practically insulated the boiler, a practice which may be followed now with advantage to both ship and boiler. A strip of asbestos, $\frac{1}{4}$ to $\frac{1}{2}$ inch thick, interposed between the boiler and its saddle, is a good arrangement for preventing thermo-electric action on the ship's frames, tank top, etc., as well as for retaining much heat in the boiler. The boiler bottom in steel ships should be well above the bilges, and room provided for a man to get in to paint or repair; the boiler bottom and ship's framework can then, and should, be kept well painted. The boiler bottom should also have a good coat of non-conduction material applied to it in such a manner as to prevent radiation, and at the same time such as to permit of easy removal and replacement for examination and repairs to the shaft.

Copper Pipes.—The whole of the pipes subject to internal pressure should be of steel, wrought iron, or strong copper; the exhaust pipes may be, and usually are in the mercantile marine, of cast iron. The Admiralty require that all pipes of steel or copper up to 6 inches diameter shall be *solid drawn.* Wrought-iron and steel pipes are now very generally employed, especially for large sizes and high pressure. If of steel and welded the Board of Trade used to require a strap riveted over the weld. Solid-drawn pipes may be obtained up to 24 inches diameter in steel of any reasonable thickness.*

The thickness of the main steam pipe should when of solid-drawn steel
= 0·125 + (diameter of bore × pressure ÷ 15,000).

The thickness of feed pipes
= 0·125 + (diameter of bore × pressure ÷ 8,000) for copper;
10,000 for steel.

The thickness of blow-off and scum pipes
= 0·125 + (diameter of bore × pressure ÷ 9,000).

When made of steel or iron welded the thickness should be 20 per cent. more than given by the above.

The thickness of copper or brass main inlet pipes
= 0·1 + (diameter ÷ 300).

The thickness of copper or brass main discharge pipes from reciprocating pump = 0·1 + (diameter ÷ 200).

If for a centrifugal pump the discharge may be of the same thickness as the inlet pipes.

The thickness of copper feed suction pipes and bilge-discharge pipes
= 0·09 + (diameter ÷ 200).

The thickness of copper waste steam pipes
= 0·05 + (diameter ÷ 500).

The flanges for copper or brass pipes should always be of tough brass, and of a thickness equal to 4 times that of the pipe; the breadth of flange should be $2\frac{1}{4}$ times the diameter of the bolts used. For pipes exposed to a pressure of 30 lbs. and upwards, the pitch of the bolts should not exceed 5 times their diameter, or 5 times the thickness of flange; their diameter is usually about the same as the thickness of flange. For pipes not subject to a pressure of more than 30 lbs., the bolts may be 6 diameters apart, or even a little more in some cases. All pipe flanges should be machined "true," especially those of the feed and steam pipes. These should fit face to face without forcing, and be jointed with as little putty or paint as possible. Brass flanges on the copper pipes may with advantage have grooves into which is fitted a ring of copper wire soft enough to easily conform to the grooves when pressed by the bolts. This can also be done with iron flanges, but the Admiralty complain that severe corrosion has taken place with them so done. Corrugated rings of soft copper or brass have also proved good jointers when fitted inside the line of bolt holes. The Admiralty require the whole range of piping to be tested by water and then by steam. The jointing that does best for water is not generally so good for resisting steam and heat. The bolting should be close, rather than wider apart with larger bolts.

Starting and Reversing Engines.—It is most important that an engine shall be capable of having its motion instantly reversed; in fact, no engine is satisfactory which cannot be stopped and made to move full speed astern in less than 30 seconds from the time of the engineer commencing the evolution. This can scarcely be done by the hand-gear with even comparatively

* The Chesterfield Tube Company can supply solid-drawn steel tubes 24 inches diameter and up to 20 feet long.

small engines, and is beyond possibility with large ones. It is essential, therefore, to provide mechanical means for this purpose in all engines of over 100 N.H.P., if such efficiency is required; and it should be of such power as to perform the operation without shutting off steam. The simplest method is to fit a steam cylinder with rods, etc., to a lever on the weigh-shaft in addition to the ordinary hand-gear, the piston pushing and pulling as required to help the engineer. There is, however, the objection to this, that the cylinder is then somewhat large, and at times the steam-power masters the hand-power, and overruns its limit, thereby tending to cause damage. This latter objection, however, is easily got over in many ways, the best of which is by adding a second small cylinder containing water or oil, which is forced by its piston from one end to the other, and thereby acts as a brake to the gear.

Brown's Patent Reversing Gear.—This idea of the brake cylinder has been worked out, and perfected by Messrs. Brown, of Edinburgh, who make a gear which, by the action of a small lever easily moved by one hand, operates on the valve motion instantly, and only to the exact extent intended by the operator, so that if the engineer moves the lever through one-quarter of its angular movement, the link-motion is moved by the gear through exactly one-quarter of its traverse. This is effected by means of a system of compensating levers, so arranged that the gear, by moving, replaces the valves in the exact position from which they were displaced by the hand-lever. Messrs. Brown have also devised an arrangement whereby the hunting and compensating levers with their rods are on the engine and so the apparatus is self-contained. Figs. 265, 265*a*, and 265*b* show the methods adopted by the firm to make the gear automatic.

Steam Gear for Reversing.—The simplest and most efficient of the steam gears, and one whose cost is so small that it may be fitted to the cheapest of engines, consists of a small engine, on whose crank-shaft is a worm, which works in a worm-wheel capable of turning freely on a fixed gudgeon on the engine frame or other convenient place; on this worm-wheel is a stud or crank-pin, to which is fitted a rod connecting it to a lever on the weigh-shaft; the eccentricity of the crank-pin is equal to half the chord of the arc through which the lever end works. The engine being set in motion, the worm-wheel is caused to revolve, and the motion of the crank-pin causes the weight-shaft to oscillate, and so to reverse the links. The steam cylinder is sometimes fitted with reversing gear, but it is quite unnecessary, and is really better without it, as the little engine moves so fast that practically no time is lost in making a complete revolution of the worm-wheel.

The advantage of this gear over others, besides its cheapness, is its simplicity, safety, and capability of being used to turn the engines when in port, or to work a winch for lifting weights when overhauling the engines; the hand-wheel is also in this case on the little engine shaft, and acts as a flywheel. The Admiralty engines have a similar starting-gear, but fitted with two cylinders whose cranks are at right angles; the starting hand-wheel is only connected to the crank-shaft or gear when required to be worked by hand; this permits of the engine starting from any position without help and stopping at any point without a brake, as is often necessary in the mercantile marine (fig. 32).

Brown's Patent Steam Starting Gears.

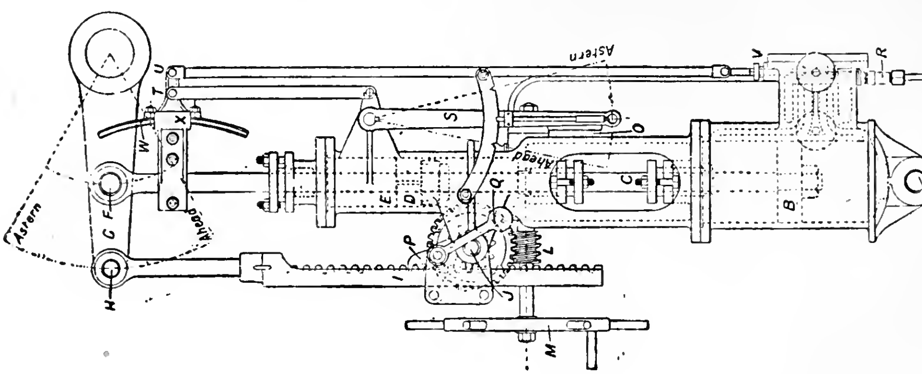


Fig. 265b.—Latest Design.

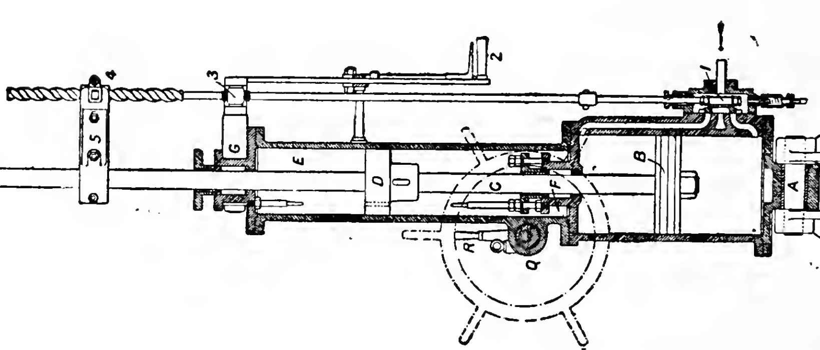


Fig. 265a.—Improved with Ordinary Hand-Gear.

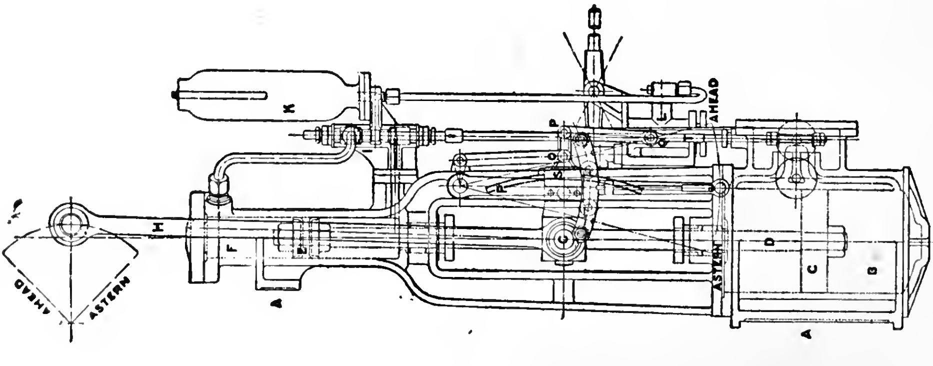


Fig. 265.—Original Plan with manual control by a pump.

TABLE LXXXII.—PARTICULARS OF ALL-ROUND REVERSING GEARS, STEAM AND HAND-MOVED.

Ship.	Cylinders and Steam of Main Engines.		Reversing Engine Cylinders.			Diameter of Worm.	Worm-Wheel.		Hand-Wheel.		Weigh Shaft.
			No.	Diameter.	Stroke.		Diameter.	No. of Teeth.	Diameter.	No. of Turns.	
	Inches.	Lbs.		Ins.	Ins.	Ins.	Ins.		Ins.		Ins.
H.M.S. M.,	40-58-88 / 54	130	1	9⅜	8	7·0	23	41	54	20·5	6¼
H.M.S. N.,	36-57-78 / 42	130	2	5	5	5¼	23	41	36	20·5	6
S.S. C., .	25½-39-60 / 39	150	2	4½	4½	5¼	23	41	39	20·5	4¾
H.M.S. E.,	40-59-88 / 48	155	2	6½	6½	6	26¾	42	54	21·0	8
H.M.S. A.,	33½-49-74 / 39	155	2	6	5	5¼	23	41	42	20·5	6·5
H.M.S. P.,	30½-45-68 / 33	155	2	5	5	5¼	23	41	42	20·5	5½
H.M.S. B.,	21-31-45 / 24	155	2	4	4	4 7/16	16¼	41	36	20·5	4⅝
H.M.S. S.,	19½-28½-43 / 18	180	1	4	3	3	9½	24	24	H 14·0 S 20·5	3¾
H.M.S. P.,	20½-33-54 / 27	250	2	4	5	4½	14¾	30	42	H 15 S 60	6
S.S. O.,	22½-32-45-64 / 42	250	1	6	6	5⅝	18⅛	38	42	19	5
S.S. A.,	32-48-80 / 48	160	1	7	8	5½	23	32	42	16	5¾
S.S. E.,	28-43-70 / 39	155	1	6½	8	5¼	23	32	36	16	4½
S.S. L.,	24-39-64 / 33	160	1	6	6	5	18	38	36	19	4¼
S.S. R.,	22-33-58 / 36	160	1	5½	6	5	18	38	36	19	4
S.S. V.,	23½-85-57 / 33	150	1	5	6	5	14½	30	36	15	3¾
S.S. D.,	18-27-48 / 27	150	1	4	5	4	12	30	30	15	3½

The Reversing of Propellers driven by Turbines is usually effected by means of a separate, small, additional turbine in rear of, and on the same shaft with, the main or ahead-going one. The astern-going turbines are generally collectively capable of developing S.H.P. equal to 25 to 30 per cent. of that at full-speed ahead-going; in Germany their power is now equal to 50 per cent. of the full-speed power. If the turbine is complete itself, or is the low-pressure member of a compound system, this stern-going instrument may be, and generally is, in the same casing with it, as shown in figs. 58 and 59. The high-pressure member of a compound system has the astern-going turbine in a separate casing in rear of it, and arranged to exhaust to the condenser direct. In this case there is required a pass valve or two valves geared so that steam may be admitted to either one of them while shut off from the other, or both may be shut off at the same time. For this purpose on ordinary slide valve, such as fitted to the

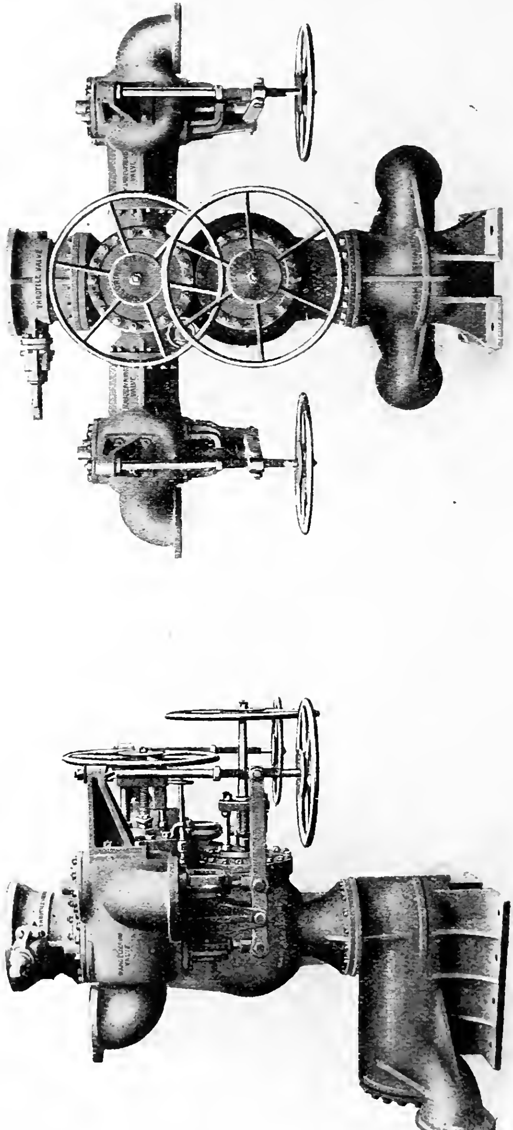


Fig. 266.—Complete Set of Manœuvring Valves for Twin-Screw Turbines.

cylinder of a reciprocating engine, or, as usual, for alternating in a steering engine, will answer the purpose; or for large sizes two mushroom valves arranged that both may be shut or one lifted without disturbing the other.

With three propellers the middle one is driven by the H.P. member, and each wing propeller by one of the two L.P. members, and as manœuvring is usually done by the wing screws only, they have reversing turbines, as in fig. 58. But in this case the main or ahead-going turbine on service obtain their supply of steam from the H.P. member driving the centre shaft, consequently in manœuvring them an auxiliary supply from the boiler is necessary for both ahead and astern-going turbines. They may, therefore, have manœuvring valves for distribution of this steam, as already described, and the middle turbine remains stopped and out of action during the "backing and filling" of the others.

Blenkinsop's Arrangement for Manœuvring is due to the observation of the late Mr. Blenkinsop, of the Great Eastern Railway Company, that with the centre screw moving in head gear the ship was much more easily handled, inasmuch as the rudder then lent powerful aid in turning under the action of the wing screws. It need hardly be said, however, that it sometimes happens that in executing a turning movement no headway must be made, but, generally speaking, this seldom is rigidly necessary for in approaching and leaving a berth a steamer may with advantage advance, and such advance under any circumstances can be provided for.

Fig. 266 shows Mr. Blenkinsop's invention whereby with

The Manœuvring Valves for Twin-Screw Turbines, shown in Fig. 266, are the invention of Mr. Hamilton Gibson and not Mr. Blenkinsop, as stated on p. 701.

... whereby the exhaust from the H.P. turbine is shut off, and boiler steam admitted to the L.P. turbine, or the reversing turbine alternately, or to neither as required. Very large valves are necessary for such work, and the gear for working them must be efficient and powerful for rapid handling. In ships of comparatively small power these valves are so large and heavy that steam-worked gearing is necessary for such handling, while in those of large power only such instruments as Brown's push and pull reversing gears are sufficient for the purpose. Fig. 267 shows the arrangement of valves and gear fitted in H.M.S. "Lion" of 80,000 S.H.P., whose power is developed in two sets of turbines, which drive four screws in the usual way.

Regulating and Stop Valves.—As it is necessary that a marine engine when required may run at any number of revolutions from full speed to dead slow, means must be provided for doing this efficiently, especially for warships whose speed when in fleet formation must be regulated to a nicety. In the old days speed was decreased by means of a throttle valve, or by "notching" up the link motion. The latter method is still adopted with advantage, but it is not sufficient when very low speeds or minute changes in speed are necessary. The old Butterfly throttle valve is no longer used, except for governing purposes, for against the high pressures now obtaining it is never tight enough for practical purposes.

them while shut off from the other, or both may be shut off at the same time. For this purpose on ordinary slide valve, such as fitted to the

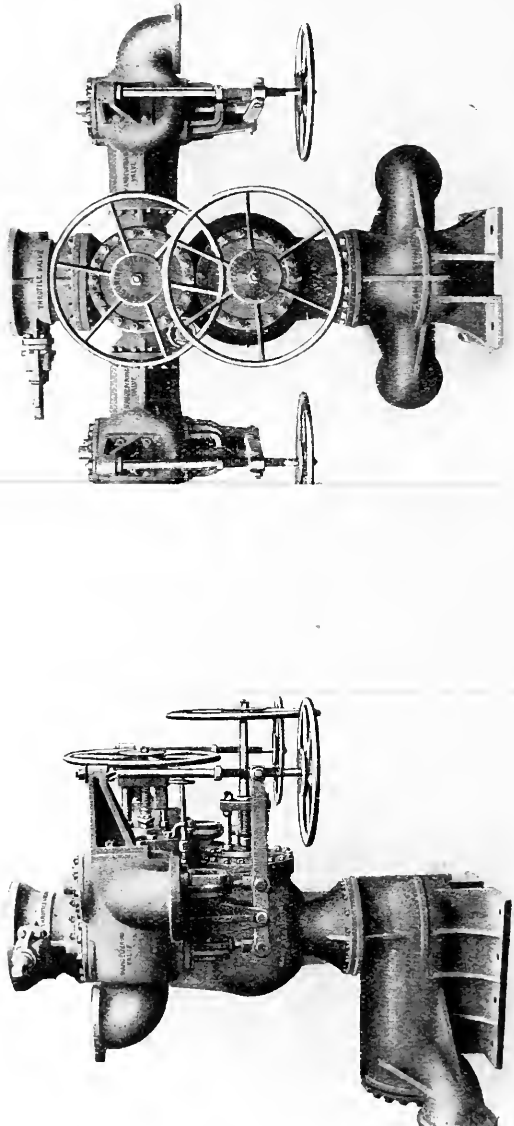


Fig. 266.—Complete Set s for Twin-Screw Turbines.

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Fig. 266 shows Mr. Blenkinsop's invention, whereby either wing screw is moved ahead or astern while the other does the reverse—viz., astern or ahead—both turbines taking steam from the centre or high-pressure one.

Ships with Four Screws driven by Turbines usually have one H.P. and one L.P. machine on each side, the L.P. ones being outermost, and having each a reversing turbine on the same casing, as already described. In this case manœuvring is done entirely with the outer screws, and all that is essential is means whereby the exhaust from the H.P. turbine is shut off, and boiler steam admitted to the L.P. turbine, or the reversing turbine alternately, or to neither as required. Very large valves are necessary for such work, and the gear for working them must be efficient and powerful for rapid handling. In ships of comparatively small power these valves are so large and heavy that steam-worked gearing is necessary for such handling, while in those of large power only such instruments as Brown's push and pull reversing gears are sufficient for the purpose. Fig. 267 shows the arrangement of valves and gear fitted in H.M.S. "Lion" of 80,000 S.H.P., whose power is developed in two sets of turbines, which drive four screws in the usual way.

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Fig. 268 shows a form of stop regulating valve which has been used successfully in the mercantile marine for many years. The little valve on the spindle is easily opened, and is sufficiently large to supply the necessary steam for manœuvring purposes if it is about one-third the diameter of the main valve. Large valves of this kind, however, offer considerable resistance

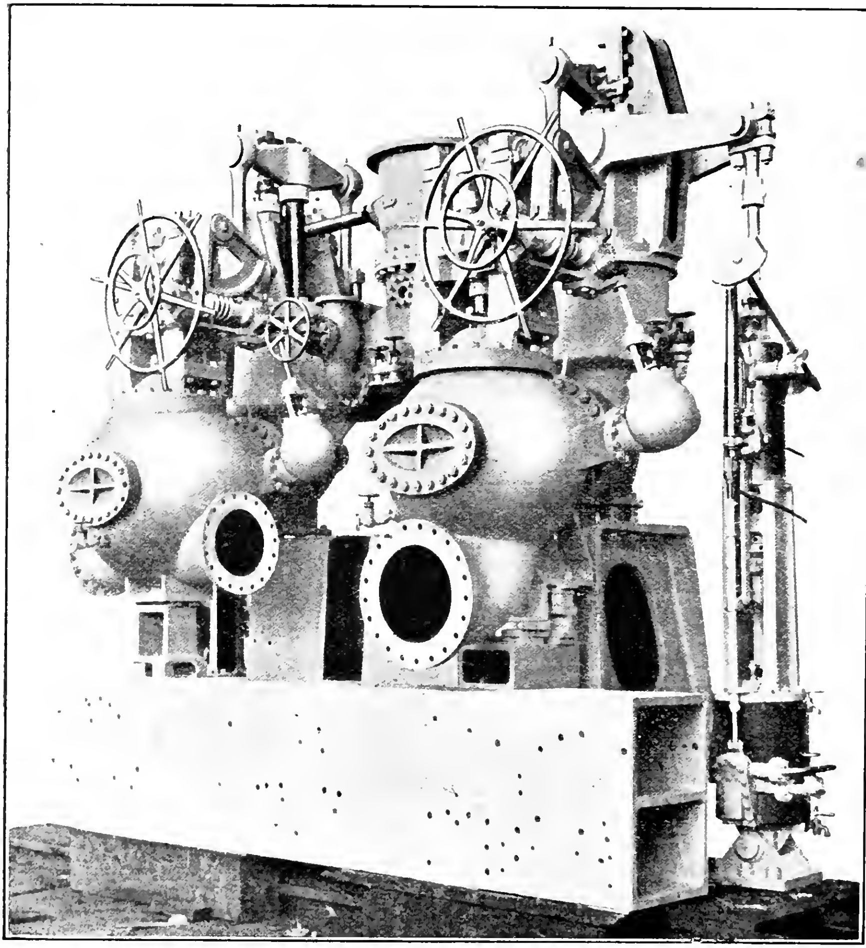


Fig. 267.—Manœuvring Gear, consisting of ahead and astern-manœuvring valve in front, with the intermediate valve behind, worked by means of a direct-acting steam engine.

to opening, notwithstanding having a pass valve, and there are other objections, so that many engineers prefer to have some form of balance stop valve, such as is shown in fig. 269, with an independent manœuvring valve on the side.

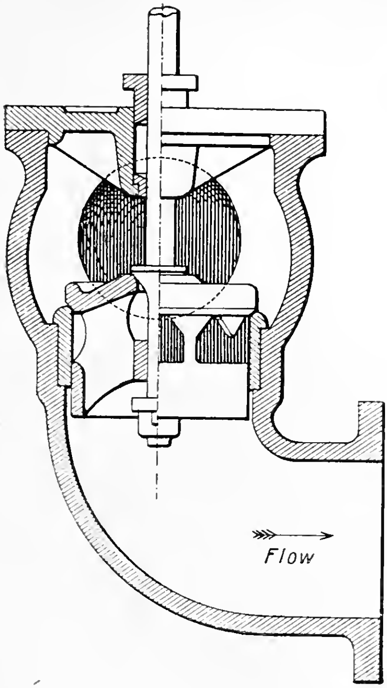


Fig. 268.—Engine Stop and Regulating Valve (for small engines).

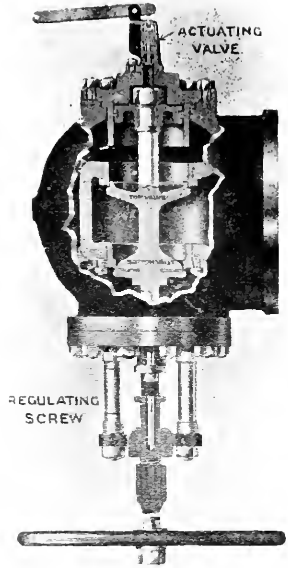


Fig. 270.—Cockburn's Double-Beat Type Regulating Valve.

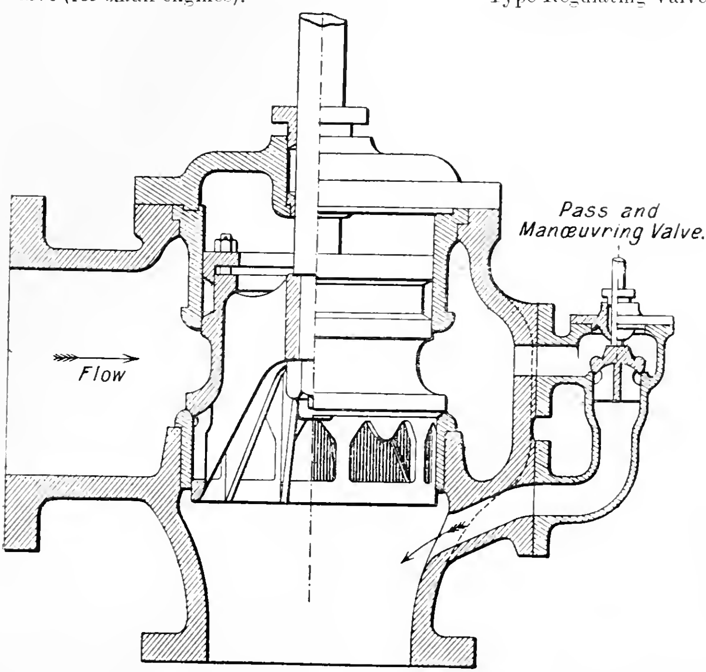


Fig. 269.—Balanced Stop and Regulating Valve (for large engines)

Fig. 270 is a form of equilibrium stop valve, self-adjusting, so that it keeps tight under differing pressures and temperatures.

The regulating valves, when properly balanced, can be worked by a lever, such as was used in the throttle valve, or with a wheel at the starting platform in the ordinary way. Very large valves, however, are too cumbersome to be easily and quickly worked by hand, and may, therefore, have

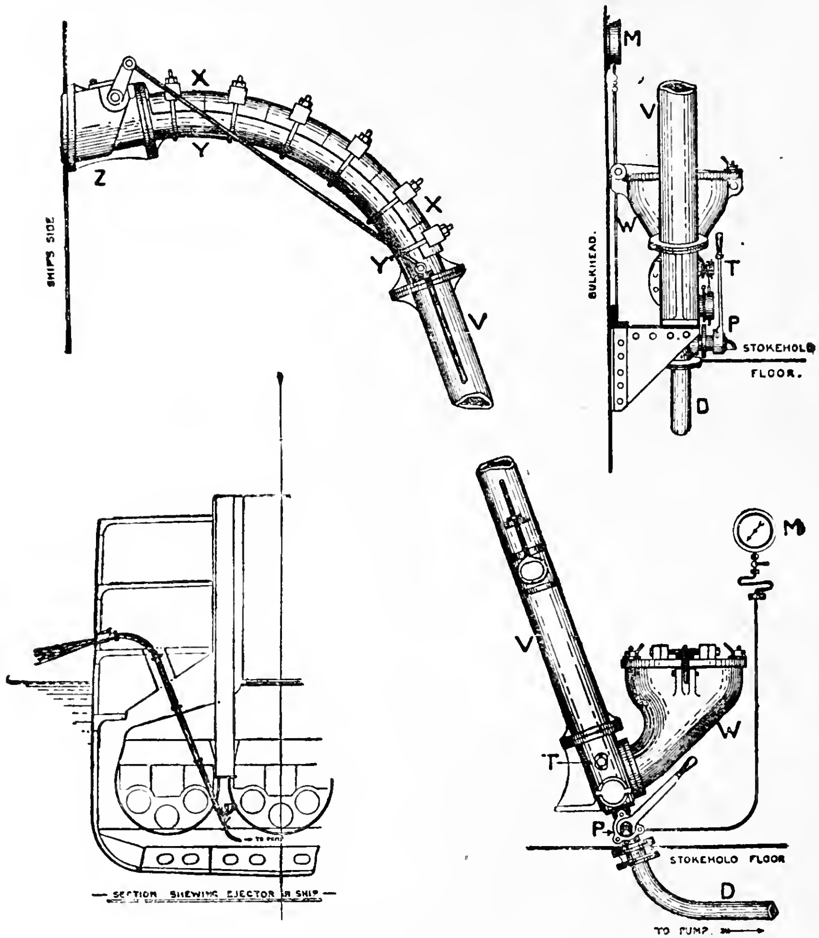


Fig. 271.—Ash Ejector.

an apparatus something like the Brown steam starting gear attached to them, and operated by a small lever at the starting platform.

Steam Turning Gear.—Another labour-saving appliance now universal in the Navy and mercantile marine, is steam gear for turning the engines when in port. In all but small ships a separate engine is provided for the purpose, but in small ships the donkey pump engine, or the reversing engine, when there is one, is employed by using belt or rope gear. The usual plan is to fit a second worm-wheel to the worm-shaft, and to turn it by a worm on the shaft of the special engine, or on a shaft with a pulley to be worked by an auxiliary engine.

Steam Ash Hoists.—All large ships require some mechanical means of disposing of the ashes, clinker, etc.; the simplest form of hoist is only a small winch worked by a steam engine without wheel-gearing; the barrel is of small diameter, and the flywheel is heavy, and kept running at a constant speed, the bucket of ashes being "whipped" up in the same way that light cargo is got out of the holds with a steam winch. This gear has the merit of cheapness and simplicity, it can be worked by the most ignorant, and does not easily get out of order.

Ash Hoist (See's), fig. 271, is an ingenious arrangement whereby the ashes are driven to the top of the tube, tilted over, and shot through the ship's side. With this gear, the fireman is not required to leave the stokehole, and no labour is required beyond that of shovelling the ashes into the hopper. Fig. 271 shows the arrangement to consist of a hopper having a hinged water-tight cover; at its bottom is a special form of nozzle, which discharges water up the delivery pipe from a special duplex pump, when the pressure has been got up in it to 200 lbs. As soon as the gauge shows the pressure, the cock is quickly opened and a rush of water up the pipe takes place, carrying with it the ashes in the hopper; the pressure drops to 150 lbs., but the stream still flows through the pipe carrying air and ashes with it as fast as the latter are shovelled into the hopper. When the whole are disposed of, the cock is closed *quickly* to prevent back wash; the hopper is closed, as is also the discharge valve on the ship's side. This apparatus is a great convenience and comfort, especially in bad weather, and saves a huge amount of unpleasant labour; it also prevents the need of the grimy and unwashed denizens of the stokehole showing themselves on the decks of yachts or passenger vessels. On the other hand, unless care is taken in placing the discharge the detritus is liable to get into the condensers with the cooling water and damage the tubes; and, further, if the cinders and clinkers are allowed to get into the screw race, injury will be done to the propeller blades, especially if of bronze.

Governors.—To prevent the engines from racing when the sea is rough, it was usual to fit an instrument which should control the throttle-valve automatically. The governor for shipboard must be designed and arranged so that the pitching and rolling of the ship do not prejudice it, but it should act rather in anticipation of the former.

There are two distinct classes of marine governor—viz., those whose action is influenced by the variation of motion of the engine itself, and those whose action is caused by the change of pressure at the stern, due to variation of head of water. The former class can only act *after* a variation of speed has taken place, the latter anticipates and checks such variations. At first sight the latter present the most favourable qualities, inasmuch as they *anticipate* change of velocity, but they serve only one purpose, that of checking *racing* of the engine due to the propeller emerging from the water. The other governors are necessarily a little late in action, but they may be made so sensitive as to be almost as quick as the others; they have, however, one superior merit, and that is, they check racing from any and every cause. If a propeller or shaft break, the pneumatic or hydrostatic governor fails to check the engine; the other governors will check any large increase in velocity and give the engineer time to shut off steam.

Both classes of governor are susceptible of subdivision, and may be distinguished as those which act direct on the throttle-valve, and those which act on the valve of a steam cylinder whose piston operates on the throttle-valve. They are, however, seldom fitted now, as the necessity for them has ceased to be acute, owing to the smaller screws with better submersion, and the multiple-crank compound engines do not race so violently as the old simple condensing. Turbines, however, do require governors for their own safety.

Dunlop's Governor.—This instrument consists of a vertical cylinder placed close to the stern of the ship, and as low down below the water line as convenient; there is a communication between the sea and the bottom of this cylinder by a cock or valve of ample size, fitted to the skin of the ship. When the stern of the ship is lifted out of the water, the cylinder is emptied of water, and the pressure in it is that of the atmosphere; on the stern dipping deep into the water again, the water rushes into the cylinder and compresses the air in it till there is a pressure due to the "head" of water, which may amount to as much as 12 lbs. per square inch in large ships, and 5 lbs. even in small ones. The top of the cylinder communicates by means of a pipe with a vessel in the engine-room, which is closed air-tight at its top by a thin corrugated circular diaphragm. The bottom of this vessel is bell-shaped, so that at its top it is of considerable diameter, and the diaphragm capable of exerting considerable force when even so small a pressure as 1 lb. per square inch is transmitted from the cylinder. The gear for operating on the throttle-valve is connected to the middle of the diaphragm, and any bulging of it by the pressure causes the throttle-valve to be opened, and when the water cylinder is emptied by the rising of the ship's stern relatively to the sea, the diaphragm assumes its normal position, and the throttle-valve is closed.

Steam Governors.—The chief cause of failure of the first marine governors was their inability to move the throttle-valve promptly when the spindle-gland was tightly packed; it was also difficult to set them so as to act effectively when they would move at all. The difficulty was got over by limiting the function of the governor to move the slide-valve of a small steam cylinder whose piston performs the operation of opening and shutting the throttle-valve. The governor is of ample power to move the small slide-valve with precision, and the steam cylinder can always be made of sufficient size to work the throttle-valve, however tightly its gland is packed. For the purpose of working the small slide-valve, a Silver's or a Meriton's governor may be employed, and many of the old governors of this kind were converted to steam ones by the addition of this steam cylinder.

Durham's and Churchill's Velometer.—The motion of the engine is communicated to this governor by a small rope of wire or Manilla on the usual pulleys; it is transmitted from the pulley-shaft to the shaft of a paddle-wheel enclosed in a cylindrical trough, by means of a bevel-wheel, whose axle is free to move about the axis of the shafts in a plane perpendicular to it, and which gears into a similar bevel-wheel on the end of each shaft. (This is similar to the arrangement provided in traction engines to admit of their going around a curve.)

The trough is filled with water or oil, which is carried around with the paddle-wheel, and causes it to resist any sudden changes of motion. The axle of the intermediate bevel-wheel is connected to the small valve of the steam cylinder whose piston operates on the throttle-valve. If the engine races so that the bevel-wheel on the pulley-shaft moves faster than that on the paddle-wheel axle, it will carry the intermediate pinion along with it, until the motion of the paddle-wheel is accelerated by it, and the pinion axle acting on the small slide-valve causes the throttle-valve to be closed. The paddle-wheel is now moving at a higher rate than its normal speed, so that when the engine has slowed down to its normal speed, the bevel-wheel axle is moved in the opposite direction, so as to cause the throttle-valve to be opened, and the engine is thus prevented from further "slowing down."

This governor, when carefully adjusted, is most sensitive, and will prevent any dangerous racing under the most trying circumstances.

Coutt's and Adamson's Governor.—This is an extension of Dunlop's principle, and differs from it by the diaphragm being caused to move the slide-valve of a small steam cylinder whose piston operates on the throttle-valve. This has the advantages and disadvantages of Dunlop's governor, except that the work of moving the throttle-valve is done by steam.

Westinghouse Governor.—A sensitive ball governor, of very small size, is made to operate on a valve which, in opening, allows the steam on one side of the piston of a steam cylinder to escape to the condenser, that on the other side forcing the piston to move quickly and shut the throttle-valve; the valve closes again as soon as the steam has escaped, and the steam flowing into the cylinder allows the piston to go back to its original position.

The piston here spoken of is connected to a smaller piston in another cylinder, whose function is to bring it back to its initial position.

This governor is very sensitive and acts very well, but is liable, from want of attention, to get out of order, and then fail to act at all. Fig. 272 is Murdoch's, and very similar and general to the Westinghouse.

With the advent of triple engines and twin screws the absolute necessity for a governor disappeared; and to-day, although some racing takes place with all ocean-going marine engines, hardly a single modern ship having reciprocators is furnished with a governor, and when such an instrument has been provided, it generally supplies ample evidence of its non-use if not of its uselessness.

Gauges.—In every engine-room there should be a steam gauge, which shall show the pressure *at the high-pressure cylinder valve-box;* there should be a gauge which shall show the pressure in each receiver (when the engine is compound); that on the low-pressure is called a *compound* gauge, as it is marked as a *pressure* gauge when above atmospheric pressure, and as a *vacuum* gauge when below—this gauge might, with advantage, be marked so as to show *absolute* pressure; if this were so an additional stumbling-block would be removed from the path of some young engineers, and permit them to have less hazy views on the question of "vacuum"; and a gauge, commonly called the vacuum gauge, which shall show the pressure in the condenser. This gauge is marked in inches, and so indicates how high the column of mercury would be in a vertical tube whose upper end is connected with the condenser, and the lower open to the atmosphere. The old original vacuum gauge was simply like a barometer, and, when replaced by the Bourdon gauge, to avoid confusion the new ones were marked in this way. To say that there is a vacuum of 12 inches, means that the difference between the pressure in the condenser and that of the atmosphere is equal to the weight of 12 cubic inches of mercury, or due to a "head" of 12 inches of mercury, or 6 lbs. very nearly. The actual pressure in the condenser is, however, not affected in the least by the pressure of the atmosphere, and gauges can now be obtained which give the real actual pressure.

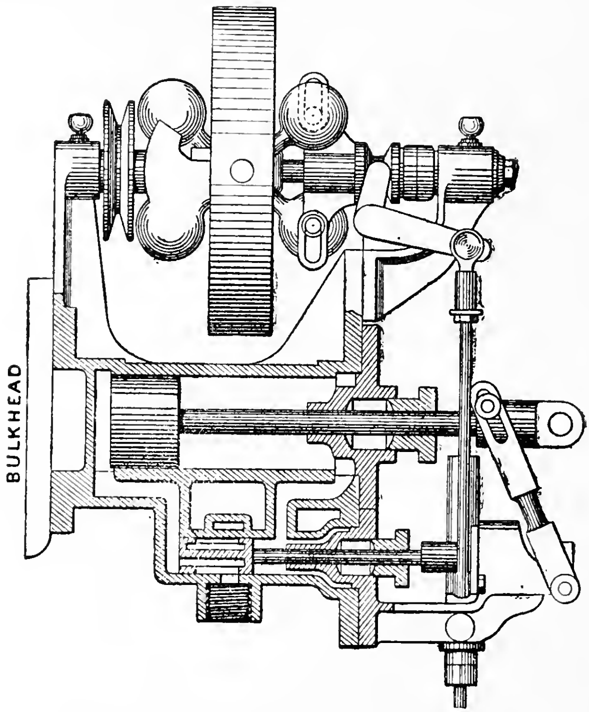


Fig. 272.—Combination Governor (Murdoch's Patent).

It would be far simpler, and certainly more scientific, to have the condenser gauge marked from 0 to 15 lbs. *absolute;* the "compound" gauge, or that attached to the valve-box of the low-pressure cylinder, marked from 0 to 50 lbs. *absolute;* and that attached to the valve-box of the other cylinders to 250 lbs. *absolute*, or to such limit as shall be at least 25 per cent. higher than the working pressure.

The gauges on the boilers might be marked as at present—viz., to indicate the pressure above that of the atmosphere, as it might be more difficult to train firemen to know the meaning of the new markings, but after all it is really immaterial to them how it is graduated, so long as they know to what mark they must keep the pointer when under steam, and that, when the pointer begins to move, on getting up steam, pressure is forming.

Care should be taken in setting the gauges in the engine-room that allowance is made for the extra pressure due to the "head" of water in the gauge pipe, which will be about 1 lb. for every $2\frac{1}{4}$ feet of vertical fall.

Lubricators and Impermeators.—To obtain perfect lubrication the supply should be steady, uniform, and continuous, or nearly so. This is true for every bearing, guide, etc., and equally true for the lubrication of the internal parts. It is usual to rely on capillary attraction to convey the oil from the oil boxes to bearings, guides, etc., by means of worsted syphons; this is a very simple, well tried, and fairly efficient method, but it has serious drawbacks; it requires constant attention, as the worsted wick syphons act as filters and become clogged with gluey matter contained in some oils, and there is no definite means of *proving* that the oil is passing. An equally simple and very satisfactory plan (fig. 273) is to fit an oil box a few inches above each bearing, in such a way that if oil is dropping from it, it can be seen or felt; there is a small cup to each oil hole leading to the bearing, and over each of these is a small nozzle from the bottom of the oil box, fitted with a small plug so as to regulate the flow of oil, or to stop it altogether; if preferred, however, syphons may be fitted instead of screw plugs, as in either case the *flow of oil can be proved*. A perfect lubrication, however, can only be got by forcing the oil by means of a pump through every joint and bearing, collecting the oil as it drops, cooling it, and again pumping it to the joints. This system of forced lubrication introduced by Messrs. Belliss & Morcom, and applied to their high-speed electric light and power engines, has proved a great success by permitting of a mechanical efficiency of 93 per cent. with even small engines, and preventing wear of bearings and journals during quite long periods of work.

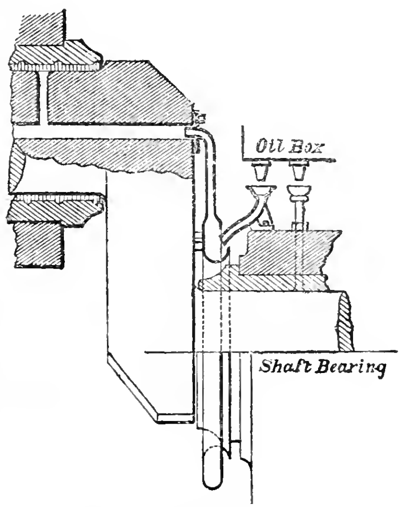


Fig. 273.—Centrifugal Lubricator.

Cadman's Patent Lubricators.—*Over* each moving part required to be lubricated is an oil box having, projecting through the bottom, a small plug, held there by a spring, and so set that the oil box on the moving part touches the plug end and opens it so as to let a drop of oil pass. This is especially adapted for the piston and connecting-rod brasses of a vertical engine, and for guides, etc.

Centrifugal Lubricators.—When engines are running at a high speed, ordinary external appliances, such as telescopic pipes, jointed pipes, pipes fixed to the rod and taking oil from wipers, etc., are not reliable or sufficient for the purposes of thoroughly lubricating the crank-pin brasses. If a system of forced lubrication by means of holes in the crank-shaft, its arm and pin, is not adopted, it is usual to have what is known as a centrifugal lubricator, such as is shown in fig. 273, whereby the oil supplied from a fixed lubricator passes through a tube, drops into a circular collar surrounding the shaft, and is whirled into its outer part and from it by means of a tube to a hole through the axis of the crank-pin, from which it flows by smaller holes to the surface and so lubricates the brasses. From the time of leaving the fixed lubricator centrifugal force is operating to compel the oil to flow to the brasses. Lengthened experience with this in the Navy has proved its reliability, and, as it is not an expensive fitting, it might be adopted with advantage generally in the mercantile marine, as less oil is wasted by it than by the ordinary appliances. The hole in the pin also provides a store of oil which would keep the brasses supplied in case of a temporary stoppage of supply from the oil-box.

Sight Feed Lubricator (fig. 274) is for the purpose of supplying a steady and definite supply of oil to the cylinders, and is so designed that the condensed steam in a pipe, etc., leading from the steam pipe, shall carry with it so many drops or globules of oil as it drains back near to the valve-box; the water condensed passes through a glass tube, near to the bottom of which the oil enters it, and the number of oil globules passing per minute may be counted, and the supply thus regulated to a nicety. This is the most ingenious contrivance, and in the hands of careful men who understand it, effects great economy, while adding to the efficiency of the engine; but in careless hands it is almost useless.

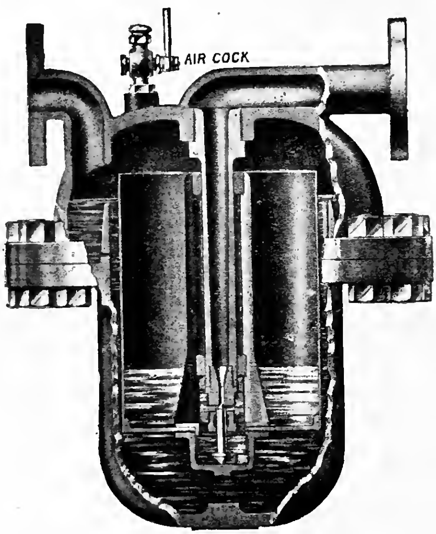


Fig. 274.—Automatic Drain from Steam Pipes or Receivers.

Mechanical Impermeators.—These are most successful in operation, and more reliable than any other form, inasmuch as their action is so simple that every one can understand them. Essentially there is only a force pump, having a very slow motion imparted to its ram or rams by gearing, moved by one of the working parts of the engine. The gearing generally consists of a ratchet lever worked from the valve-rod of the nearest cylinder, and moving a wheel on whose axle is a worm which gears into a wheel on the rim of a nut; this nut fits on the thread cut on the ram, and held in position by a bracket; as the nut is moved round, the arm moves slowly in or out of the chamber. The ram chamber is supplied with oil from a small tank, and is connected by a pipe to the top of the valve-box of the high-pressure cylinder: small non-return valves are fitted, so that the oil cannot flow back to the tank, and the steam cannot force itself or the oil back into the chamber. It is, however, a very common practice now to run engines without internal lubrication, or with only such as enters the cylinders with the piston and valve-rods. It has been found that after an engine has worn its internal moving parts fairly smooth there is no need of other lubricant than the moisture from the steam, and so all risk of damage to boilers from grease is avoided by doing away with internal lubricating apparatus. If, however, superheated steam is used, oil lubrication may become necessary in some engines; when it does, care should be taken to use only a pure hydrocarbon suitable to the temperature or graphite paste forced in by a similar instrument.

Drain Pipes from the cylinders and valve-boxes should lead to the condenser, so that there shall be no loss of fresh water, and no filling of the engine-room with vapour when the cocks are opened. It is customary to connect these pipes to the hot-well, which serves this purpose very well with high-pressure and medium-pressure engines; but since the pressure in the low-pressure cylinder of a compound engine is only for a very small portion of the stroke above that of the atmosphere, the opening of the cocks will not get rid of the water, but only allow air to force its way back, and so reduce the vacuum. It is sometimes convenient to see if water is flowing, and so prove that the cocks are not choked; this may be accomplished by fitting a "three-way" cock to the main drain pipe, which permits communication to be made with the condenser or bilge at the will of the engineer.

Jacket Drains should always lead to the hot-well, and when the engine is working the cocks should be open sufficiently wide to just keep the jackets free of water, or there should be automatic drainers fitted; the hot water escaping from the jackets then helps to warm the feed-water.

Feed-heaters.—It is a most essential thing that the feed-water shall enter the boiler as warm as possible, both to obtain evaporative efficiency and avoid wear of the boiler. Economy can only be effected *directly* by making use of heat that would otherwise be wasted for this purpose; but *indirectly*, by promoting circulation and reducing the necessity of circulation instead of checking it, considerable economy may be effected, so much so indeed as to warrant the use of heat which is not "waste." Many attempts have been made to heat the feed-water with exhaust steam, hot gases in the uptake, etc.; but no great measure of success has attended the efforts of those who have paid most attention to this, for the apparatus employed has generally been inefficient and its durability short. In the older expansive engines, where the temperature of the steam at exhaust was often over 230° Fah., great economy was effected by heating the feed-water in a small kind of surface-condenser, placed on top of the condenser so as to intercept the hot current of steam flowing to the latter; but now with compound engines, where the temperature at exhaust is only about 180° Fah. at the most, no such means is efficient for the purpose. No doubt *some* economy is possible even under these circumstances, especially if the feed-water is permitted to *circulate* in the heater for an appreciable time; and considering that now, at a much lower temperature—viz., 100° F.—a little more heat can be imparted to it from an external heating agent, the temperature being 80° above it. Some day, perhaps, means will be found to avoid the loss of all the *latent* heat which takes place, and which is huge compared with the whole of the *sensible* heat. The exhaust steam being at 180° F., the total possible saving of sensible heat is now 50° F., while the latent heat lost is nearly 1,000° F.

Weir's Feed-Water Heater and Automatic Regulating Gear are designed to raise the temperature of the feed-water to nearly 212° Fah. by means of a portion of the steam from the L.P. valve-box of the compound engine. Fig. 275 gives a sectional view showing its general construction. The heating steam is taken from the low-pressure casing of the main engine, and the exhaust of the auxiliary engines, such as feed pumps, electric light, fan engines, etc., is also led into the heater through the non-return valve B on the side of the apparatus. A circular ring and conical spray piece with perforations are fitted to mix the water and steam uniformly. The feed-water is forced by the main engine feed pumps through the spring-loaded valve D on the cover in a thin sheet, and is instantly heated by contact with the steam. As the pressure in the heater is generally much less than that of the entering water, the effect of this lowering of the pressure, and sudden heating of the water, is to liberate the air in the water, and this is removed to the condenser, or to the atmosphere, by a small cock K on the air vessel placed on the top of the heater. The feed-water is thus rendered non-corrosive, and falls to the bottom of the heater at the boiling temperature due to the pressure. The steam admission valve is of special construction. It can be opened to admit the necessary amount of steam to the heater, but it closes by its own weight in case there is no flow of steam into the heater: this valve is also fitted with a dashpot, which allows the valve to close gradually, and prevents it hammering on its seat in case of fluctuations of pressure. The combination of the automatic regulating gear with the heater has long been a special feature of the Weir apparatus. The float, E, shown in the lower part of the heater is a pan, with water-tight bottom and sides, but open on the top. It is suspended on two levers, so as to move up and down with a parallel motion; the top lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance when one-half is immersed in water. To the weight lever another lever is attached, which actuates the throttle valve F and controls the supply of steam to the pump drawing from the heater. When the water in the heater rises the float is raised, and the throttle valve opened, and, when the water level is lowered, the float

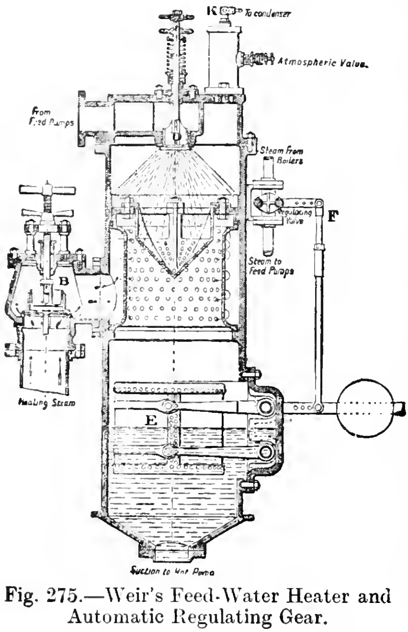


Fig. 275.—Weir's Feed-Water Heater and Automatic Regulating Gear.

follows, and the valve is closed; the level of the water is thus kept constant in the heater, and the pumps are completely filled with water. The regulating valve is a cock with a parallel key: the pressure of the steam keeps it perfectly steam-tight, although it may have worn slack in the shell; the pressure also keeps the shoulder of the key against the bottom of the stuffing-box, so that the stuffing gland is always kept slack. A relief valve and the necessary gauges are also fitted to the heater.

Evaporators or Distillers.—The advantages of supplying marine boilers

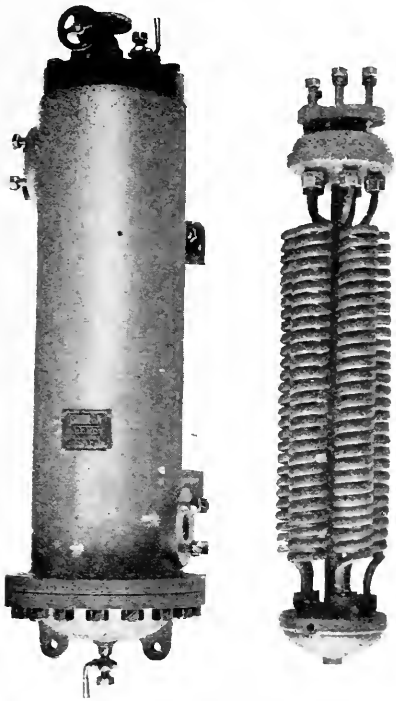


Fig. 275*a*.—Live Steam Feed-Water Heater (Caird and Rayner).

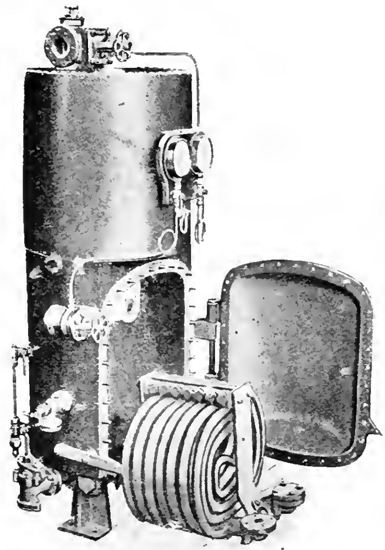


Fig. 276.—Evaporator.

with pure water are great, and are so obvious as not to need specifying. Samuel Hall, the successful introducer of the surface condenser, was so thorough in his desire to use only fresh water in the boilers of ships with his condenser that in 1833 he fitted evaporators to several of them made on precisely the same principles as govern the design of our modern ones. He, however, went beyond the modern engineer by placing his apparatus in the steam chest of the boiler itself, so as to lose no heat. The necessity of it was not, however, so severely felt until voyages of considerable length had been made with ships whose boilers work at pressures of 100 lbs. and upwards. The weight of water evaporated in boilers, whose working pressure is 150 lbs., is much greater in proportion to the size than was the case with those working at 75 lbs.; and the evils arising from the deposit of scale are magnified with the higher pressure and consequent higher temperature. Again, the liability to put on scale is greater, inasmuch as the losses from leakages are greater with the higher pressures. Hence, the old system of making up loss of water by a supply from the sea, although a very simple and ready one, was not by any means satisfactory, and did not remedy the evil, but rather magnified it. The Admiralty and some private shipowners tried to obviate it by providing a supply of fresh water in the double bottoms, or in tanks specially fitted for the purpose. This, however, was only half a remedy, inasmuch

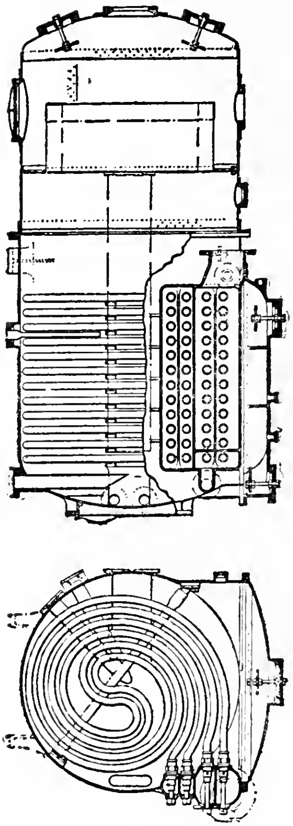


Fig. 277.—Weir Admiralty Evaporators.

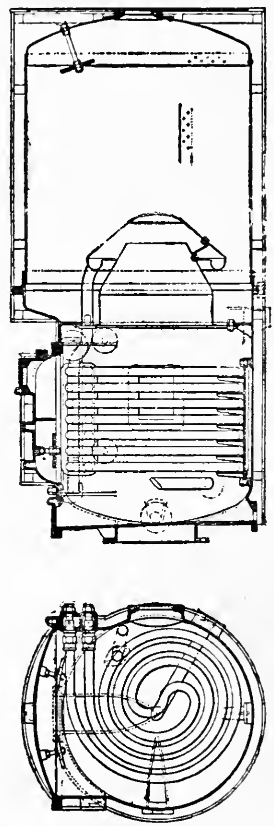


Fig 277*a*.—H.P. Evaporator for Destroyers.

as the fresh water generally obtainable contained large quantities of lime and other salts which gave a hard deposit more difficult to remove. Moreover, this fresh water cost money, and was so much extra weight to carry, which also added to its cost. Then recourse was had to the auxiliary or donkey-boiler to obtain distilled water, which meant an expenditure of coal as well as labour in cleaning out these boilers after they had become coated. Besides, these small boilers very soon got so coated that they had to be stopped for a thorough clean-out; and during the time they were at work there was always the risk of damaging them. In spite of these difficulties, however, it was found to be the most satisfactory way of obtaining an extra supply for the main boilers, and, consequently, improvement was made in this direction by designing and supplying a small boiler (fig. 276), whose heat is obtained from either the steam direct from the main boilers or from the exhaust from one or other of the cylinders; the former plan was found eventually to be the best.

Fig. 277 may be taken as another example of the type now in general use. It consists of a vertical cylindrical shell fitted with mountings and gear similar to those on a steam launch boiler; instead of a furnace, combustion chamber. tubes, etc., it has a tubulous arrangement ingeniously contrived so that the steam is made to give up its heat to the water within

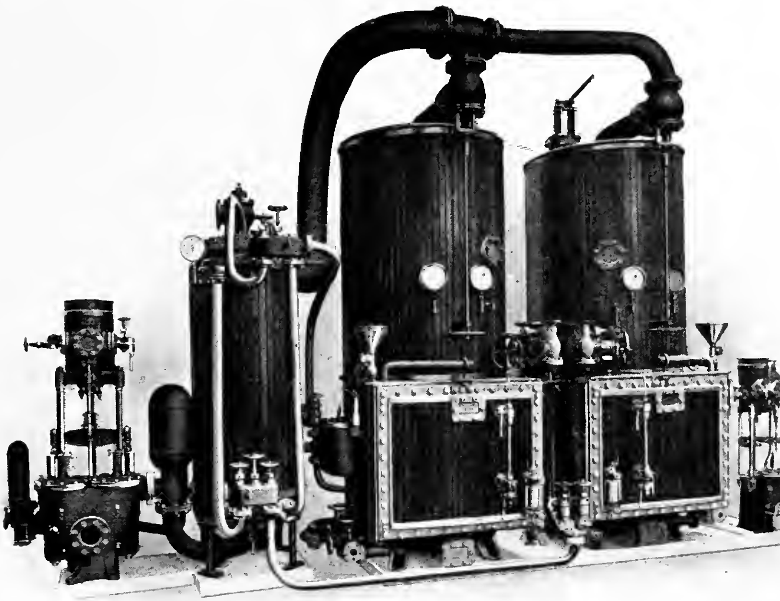


Fig. 278.—Fresh-Water Distilling Apparatus for British Battleships and Large Cruisers (Caird and Rayner).

the evaporator as far as possible, and the resultant water to drain away and be returned to the main condenser or hot-well. Steam is in this way raised in the evaporator, and passes from it to the main condenser or, as in naval ships. to the auxiliary condenser, the resultant water being finally pumped into the main boilers in the usual way. Sea water is pumped into the evaporator by a small donkey-pump, and the salt is blown down from the evaporator in the same way as was usual with boilers supplied with sea water. The internal tubulous apparatus is so arranged that it can be easily withdrawn from the shell, as shown in the figure, for a thorough clean-out when necessary. Instead of the steam from the evaporator being sent direct to the condenser, it can be made to do useful work by admitting it to the valve box of the low-pressure cylinder.

There are other equally ingenious and efficient evaporators, but they are all worked on the same principle of heating water and converting it into steam with steam made in the main boilers. In all cases, as is only to be expected, tubes are employed in one form or another to effect this purpose.

Ladders.—The main ladder to an engine-room should not be less than 18 ins. wide. and where space permits should be 24 ins., and even more in large ships; the sides are of flat iron bars usually, $4 \times \frac{3}{8}$ in.; the treads or steps are of cast iron, and 10 ins. apart, and from $4\frac{1}{2}$ to $7\frac{1}{2}$ ins. wide.. The inclination of the ladder to the vertical is usually about 1 in 3 with narrow, and 1 in $2\frac{1}{4}$ with broad steps; the hand rail is 1 in. diameter when of iron, and from $1\frac{1}{4}$ to $1\frac{3}{4}$ ins. when of brass; the former looks better from an engineer's point of view, is more durable, and easier kept clean. Ladders leading to the various parts of the engine are usually made lighter than the main ladder,

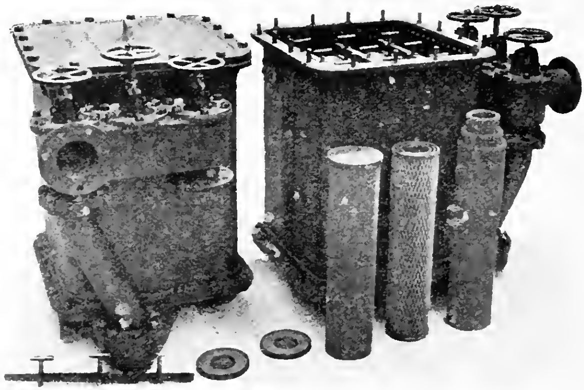


Fig. 279.—Gravitation Type Feed-Water Grease Extractor (Caird and Rayner).

and the steps are often formed of three "spills" or bars, $\frac{3}{4}$ in. diameter, or, better still, square section; when weight is important they may be of triangular section.

It is very essential that means be provided for the engineers to get easily and safely to every part of the engine requiring attention; these light ladders are a source of great convenience, and are amply paid for in the better attention given to the working parts.

Gratings and Platforms.—With the same object in view, good platforms to stand on, and gratings to form roads to the various parts should be provided. The engine-room platform is usually laid with either cast- or wrought-iron chequered plates, the pattern on whose face should be one which will give good foothold, and not prevent dirt from being swept from it. Some engineers prefer to have all bottom platforms made of wood, and laid over with sheet lead; the lead permits of good foothold, and is easily kept clean; water runs easily off it, and when in good repair it looks very well; but it is not so durable as the iron, and is very liable to damage from weights falling on it. The gangways leading to the upper parts of engines are sometimes made with chequered wrought-iron plates; but except when they are immediately over the working parts this is not a good practice, as both light and ventilation are obstructed by them, and they require constant cleaning. "Spill" gratings are, therefore, preferable in most cases, as they stop light and ventilation to only a very small extent, and require no cleaning; they are made with sides of flat bars $2\frac{3}{4} \times \frac{3}{8}$ in., and cross bars $\frac{5}{8}$ in. diameter, and spaced $2\frac{1}{2}$ ins. apart; those of large size and liable to support heavy weights have sides $3 \times \frac{7}{16}$ in., with spills $\frac{3}{4}$ in. diameter, pitched $2\frac{5}{8}$ ins.

Feed Filters are often fitted in ships of all sizes with the object of preventing any oil or grease getting into the boilers with the feed-water; they certainly succeed in keeping out grease and solid matter, but fail to do so with oils. Generally the filtering material is Turkish towelling so placed in one or more layers that all the feed-water must pass through it and leave deposited on it all solids and semi-solids, and to a great extent absorbed by it oils and greases in the liquid state. The apparatus is always arranged so that the "cartridge" of towelling can be easily withdrawn and a fresh clean one substituted without causing any stoppage of the engines (*v.* fig. 279).

Stokehole Ventilators should have an aggregate transverse area of 0·45 square inch for each pound of fuel burned per hour; or, say, 0·675 square inch per I.H.P. of trial trip in the Mercantile Marine, or 0·75 square inch per I.H.P. in Express steamers on short service and naval ships; for turbines 0·62 per S.H.P. is sufficient.

The area of the mouths should not be less than—

1·35	square inch	per pound	of fuel in	10	knot ship.
1·24	,,	,,	,,	$12\frac{1}{2}$	,,
1·13	,,	,,	,,	15	,,
1·03	,,	,,	,,	$17\frac{1}{2}$	,,
0·93	,,	,,	,,	20	,,
0·85	,,	,,	,,	$22\frac{1}{2}$	,,
0·78	,,	,,	,,	25	,,
0·72	,,	,,	,,	30	,,

The diameter of the mouth is usually twice that of the neck, but with very large down-casts the mouth is somewhat less. If d is the diameter of the neck and D that at the mouth, then—

$$D = 1{\cdot}75\, d + 5 \text{ inches.}$$

The maximum flow of air in feet per minute through the neck (d inches) should be $1{,}000\ \sqrt[4]{d}$.

CHAPTER XXVII.

WEIGHT AND OTHER PARTICULARS OF MACHINERY RELATING THERETO.

The Weight of the Engines and Boilers of every ship is a matter of extreme importance, although certain classes of shipowners rarely give a second thought to it, and probably would not pay anything extra for a saving of 5 per cent., although it would enable them to carry more cargo and gain yearly a respectable sum of money. In the case of small steamers of high speed it is of the very first importance; in fact such speeds as they are run at to-day are only attainable by the most careful designing of the engines, whereby the weight is reduced to a minimum. The best modern instances of this are to be found in the Torpedo Boat and Torpedo Boat Destroyer; some good ones are also observable in the cross-channel steamers, both paddle and twin screw. In the case of the modern Atlantic steamers the margin for weight of cargo is exceedingly small, so that a ship may have 50,000 tons displacement, and only a margin of 500 tons for cargo pure and simple. The machinery of such a ship will weigh 8,500 tons. It will be seen, therefore, that the saving of a small percentage of its weight would add a very large percentage to the dead-weight cargo capacity. (Of course there is plenty of room for stowing cargo in these ships, so that they can carry plenty of goods, whose specific gravity is low, and thereby obtain high rates of freight if there is surplus displacement sufficient.) In the case of cross-channel steamers the present speeds were only possible by the reduction in size of boiler per I.H.P., consequent on the introduction of the triple and quadruple engine and forced draught. In their case, however, it was not merely a question of weight, but one of capacity as well. In the case of the cargo steamer the weight of the machinery is only a comparatively small fraction of the total displacement, and hence is, as already stated, too generally disregarded; but here, too, the question is worthy of attention, as in the times of bad freights the difference of a few tons in weight of cargo carried may make the difference between a profit and a loss.

Most modern engineers, however, have their attention called to the question of weight by other considerations than obtain with the shipowner.

In the case of warships, weight of machinery again is of prime importance, inasmuch as the armament and fuel supply is in no small measure dependent on the margin of displacement left, after providing for the whole equipment and machinery. In other words, the lower the weight of the engines and boilers, the larger may be the armament, the armour, and the fuel supply. In the case of battle ships, whose speed is comparatively moderate, and the displacement on the dimensions, consequently larger than is the case with cruisers whose speed is very high, the machinery need scarcely be so light; and although, as a rule, the design of both engines and boilers is similar to that of the cruiser the rate of revolutions is not nearly so high.

Again, the small cruisers, whose speed is to be as high as that of the big ones, have not the same margin for their machinery; it is, therefore, imperative that their engines should be lighter per H.P. The saving in weight is generally effected in this case by making the engines to run at a very high number of revolutions, with a high effective steam pressure, and designing the various parts with the greatest care possible, using only material of the greatest strength for the purpose. The factor of safety, therefore, is practically as high as that of other ships, because these ships may have to run in war-time continuously at or near full speed for long periods.

The machinery of the Torpedo Boat and Destroyer is, of course, also designed with the greatest care, and made of the very best material, but here the so-called margin of safety is reduced; in other words, engines are made so that at their ordinary rates of speed there will be as good a margin of safety as in any other ship; spurting they will have to do in time of war, but their fuel supply does not permit of long duration of it, and the exigencies of the service in peace time would not require it to be done often in the course of a twelve-month; hence, although the factor of safety at high speeds may be low, it is still well within the safe-working limit of the material, and no great risk of accident is run. A reference to Tables lxxxv. to lxxxviii. will show the progress of modern engineering in both the Navy and mercantile marine during the past thirty years, and the lines on which modern engineers have advanced to attain their present success in coal consumption and the weight of machinery.

In 1870 Battleships and Cruisers were still fitted with simple horizontal engines and box boilers of low pressure. They were on the expansive principle, steam being cut-off at a little past half stroke, and exhausted into a surface condenser. The boilers were designed for a pressure of 30 lbs., although sometimes 1 lb., or even 2 lbs., more was attained. The maximum piston speed was about 600 feet per minute on trial trips with large engines, and about 50 to 500 feet with the smaller ones. The mean pressure was about 20 lbs. per square inch in battleships and 20 to 25 lbs. in cruisers. The I.H.P. per ton total weight machinery was about 7·0; 8·5 I.H.P. was developed for each square foot of grate, and 33 per 100 square feet of total heating surface; the consumption of coal was about $3\frac{1}{2}$ lbs. at full speed, and about $2\frac{1}{2}$ at half power.

The Battleship of 1905 was fitted with reciprocators, had the four-cylinder triple-compound engine balanced on the Schlick Tweedy system, and running at high revolutions gave a trial speed of 19 knots. To-day all British battleships and most foreign are driven by turbines operating on four screws, developing 35,000 to 60,000 S.H.P., and attaining a speed on trial of 25 knots with 330 to 380 revolutions per minute.

The Modern First-class Cruiser differs from the battleship chiefly in speed, and consequently in length. The turbine-driven ones of to-day attain a speed on trial of 33 knots, and some are said to do even more. Steam is supplied by water-tube boilers, and the revolutions at full power are about 300 per minute. The ships previous to the adoption of the turbine were of very considerable size and power, and attained speeds of 23 to 24 knots with twin screws running from 110 to 135 revolutions per minute. Their engines were generally of the four-crank triple-compound, like those of the battleship.

The Second-class Cruiser of to-day is larger and much faster than those of the first-class of ten years ago. Their speed, when driven by turbines, is 26 knots at about 500 revolutions.

The Light Cruiser and scout are modern creations, having a speed of 25 to 26 knots when driven by reciprocators at 200 to 210 revolutions; but when driven by turbines 30 knots is general, and by many later ones 36 knots has been attained.

The Destroyers are now of two classes; the one for coastal defence and attack, the other ocean-going as videttes to a fleet. The early destroyer of twenty years ago was 180 to 190 feet long, had twin-screw compound engines of about 4,000 I.H.P. at 360 to 400 revolutions; they were followed by similar but somewhat larger vessels, which attained a speed of 30 knots with triple engines indicating 6,000 to 7,000 H.P. at about 400 revolutions. A few faster ones were built of the same type, whose speed was 32 to 33 knots. To-day the destroyer for shore work has a speed of 25 to 26 knots, with a power of about 7,000, and the sea-going craft 35 knots driven by three screws and turbines of 25,000 S.H.P., running at about 750 revolutions per minute. The high speeds attained by this class of ship are due to the very light construction of both hull and machinery, but the later reciprocators were designed 30 per cent. heavier than the early ones, with which some trouble was experienced on account of lightness. The tendency to-day is to revert back to twin screws for small craft driven by compound turbines, each complete in itself, and capable of quite independent treatment, or by turbines connected to the screw shafting by helical wheel gearing or electrical generators and motors: the impulse turbine being preferred as more convenient.

Piston Speeds.—In 1870, prior to the introduction of the compound engine, the piston speed of an Atlantic passenger steamer was 400 feet per minute, of smaller passenger steamers 350 feet, while fast cruisers had a piston speed of 550 feet; battleships the same, while small cruisers had only 450 feet.

In 1880, before the advent of triple-compound engines, Atlantic and ocean steamers generally had advanced to 600 feet, the smaller passenger steamer to 450 feet, the cruiser to 600 feet, the battleship to 600 feet, and small cruisers to 550 feet per minute.

In 1882 forced draught was introduced into the Navy, and by this means the increased supply of steam permitted of higher speeds, so that in 1885 large cruisers had a piston speed as high as 660 feet, and small ones as high as 750 feet per minute. At that time the triple engine had got a foothold in the mercantile marine, and permitted of a piston speed of 750 feet in ocean mail steamers and small passenger steamers 550 feet per minute.

By 1890 higher pressures of steam had been employed, and confidence in the triple engine established; consequently the piston speed for cruisers and battleships was raised to 800 feet, ocean liners to the same speed, and small passenger steamers in some few instances exceeded this.

Five years later battleships had a piston speed of 875 feet, cruisers a little more; while the newcomer, *Destroyer*, attained its very high speed by moving the pistons at a velocity of 1,200 feet per minute. The Atlantic mail and the small passenger steamer had made only very slight advances on 1890.

In 1905 battleships and cruisers were fitted with four-crank carefully balanced engines running at 960 feet, and small cruisers at 1,000 feet per minute; the Atlantic mail steamer at 975, and the small express steamer at 950 feet. The cargo steamer pure and simple makes her voyages with a piston speed of 500 to 660, depending on the size of the engine.*

The increase in piston speed had been obtained by quickening the revolutions from 70 to 120 in battleships, and from 75 to 140 in large cruisers; the greatest difference is, however, in small cruisers, the change being from 90 to 250 revolutions. In the mercantile marine the revolutions of an ocean liner are 80 against 55, and in ordinary small passenger steamers 150 against 80 to 90 of thirty-five years ago. It is therefore not surprising to find that the I.H.P. per ton of machinery in ocean mail steamers is only 6·75 as against 5 or 35 per cent. increase, while in large cruisers it is 12 against 7, or an increase of 70 per cent. In small cruisers the I.H.P. was 19 per ton against 6 of thirty-five years ago, or an increase of 216 per cent. It may be taken as a rule that a reduction in weight per I.H.P. is obtained by increased piston speed if it is due to high revolutions; on the other hand, if piston speed is got by lengthening the stroke, no saving is effected, but, on the contrary, an increase may be looked for.

Boiler and Mean Pressures.—The general tendency of increase in boiler pressure is to increase the weight of the engines as well as the boilers. If, however, an increase of mean pressure is obtained by the increase of boiler pressure, and the rate of increase be greater than that of the boiler pressure, the weight of the engines per I.H.P. may be really decreased. Such, however, was not generally the case in the history of marine engineering progress. In 1870 with a boiler pressure of 45 lbs. absolute, the mean pressure was as much as 24 lbs. in the expansive surface condensing engines; or 53 per cent. of the initial pressure. In the compound engine of about that time the referred mean pressure was only about 22 lbs. with a boiler pressure 75 lbs. absolute, or only 29 per cent. Later, with a boiler pressure of 105 lbs. absolute, the mean in express steamers was only 30 lbs., or 28·5 per cent. Later still, with the triple engine and a boiler pressure of 170 lbs. absolute, the mean in cruisers of high speed was 38 lbs. or 22·4 per cent. and 32 or only 18·8 per cent. in merchant ships. Finally, with initial pressure of 250 lbs. in the Navy the referred mean pressure was about 20 per cent., while in the mercantile marine now with 215 lbs. it is 17 per cent.

The effect of increase of pressures is best seen by comparing the second-class cruiser of 1898 with that of 1896. These engines developed practically the same power at the same number of revolutions; the design of each was identical, but the cylinders were made smaller in diameter for the increase in mean pressure, the referred mean pressure being 50·9 lbs. in one case and 40·3 in the other. As might be supposed, the weight of engines is practically the same in each case. If, however, the boiler pressure had been increased without an increase of referred mean pressure, the I.H.P. would have been the same, but the weight of engines greater. If the boiler pressure had remained unaltered but the referred mean pressure increased, there would be an increase in I.H.P. without a corresponding increase in weight. It may therefore be assumed that increases in boiler pressure only add to the weight of the *engines* when there is no substantial increase in the referred mean pressure. Also, that when there is a substantial increase in the referred

* By the N.E. Coast Standard Specification the piston speed ranges from 448 with 3 feet stroke to 560 with 4·75 feet stroke on service of cargo boats.

mean pressure, the weight of *machinery* per I.H.P. is not necessarily increased by the increase in the boiler pressure.

Materials.—The weight of machinery is, of course, very considerably influenced by the materials of which it is made. For example, the ordinary mercantile engine is largely made of cast iron of a quality whose strength does not exceed 7 tons per square inch. The engines of naval high-speed ships have little cast iron in their construction, but what there is can withstand 11 tons per square inch, and some of it even as much as 13 tons; much of them is of cast or wrought steel, having a strength of 28 tons in place of the merchants ships' 7-ton cast iron. Again, in naval ships the bronzes are extensively used in lieu of cast iron, and although Admiralty bronze is but little stronger than best cast iron, it is much more so than the 7-ton castings, and consequently may be, and is from one-half to two-thirds the thickness of the corresponding iron castings of the mercantile marine. The light weight per I.H.P. of the cross-channel mail and passenger steamer is largely due to their having engines made of the same materials as naval engines. New materials are frequently brought to the notice of engineers by metallurgists and manufacturers which promise great things in the way of saving weight, and some of them fulfil the promise, but others. from one cause or another, disappoint both their discoverers and users. Aluminium is an instance of this, and was very disappointing till Vickers, Ltd., introduced Duralumin, which, while being very little heavier than aluminium, has a strength and toughness far in advance of it, and sold in bars and sheets at quite a moderate price; it cannot, however, be cast. Nickel steel and venadium steel, also chrome nickel, are serving the turn of the naval designer by withstanding the impact of shot; they also serve the turn of the engineer by withstanding high stresses—especially alternating ones—and any number of repetitions of such shocks as come on many of the moving parts of the marine engine.

Annealing and oil tempering, too, may be more extensively used to improve steels as now made, or at anyrate to make them more uniform in nature and with more of the valuable character as usually described by their friends than as found always in practice.

The superior bronzes, too, may be used with advantage to save weight, inasmuch as the ultimate strength of some is equal to that of steel; moreover, castings may be made of them much thinner than if of steel. Most of these bronzes can be forged and rolled, when their strength is improved, and they are rendered even more suitable for high-class engineering than in the cast state.

Design has great influence on the weight of marine machinery, as may have been supposed; and what can be done by the most careful designing is seen in the case of the machinery of the *Destroyer*, when 45 I.H.P. per ton is developed, or even in that of the third-class cruisers, when 20 I.H.P. was developed under forced draught, and even 14·5 under natural draught per ton weight of reciprocating engines and boilers.

In the case of the *Destroyer* it will, of course, be urged that the saving was effected largely at the expense of strength. This cannot be said of the cruiser's engines any more than it can be of those of the cross-channel steamer.

It is true that not much saving can be effected in the principal working parts, except by such means as boring the shafts and rods. No doubt now, when improved tool steel and appliances admit of boring at a cheap rate. hollow shafts will be used as freely in express steamers as in naval ones, and with the same satisfaction. The great saving will be effected by change in design of the fixed portions, and in most of the smaller details being dealt with drastically. It astonished many marine designers who turned from large engines to consider the quick ones in *Destroyers* to find how much material was practically wasted in the numerous unconsidered fittings and fixings of an engine and engine-room.

Not much in any case can be done in reducing the weight of the parts subject to *alternating* stresses, except by the use of materials better calculated to withstand them, such as special steel, etc., oil tempering, etc. Some saving, however, can be made on those parts liable only to *intermittent* stresses, especially when of a definite and easily calculable nature; and still more may be done in this direction on the parts subject to simple, steady loads.

Table lxxxiii. gives the safe limit of the various metals used by engineers under the several conditions obtaining in practice, and the following observations may be taken as guides in dealing with the question of design :—

Effect of various Loads and Stresses.—The various parts of marine machinery are subject to loads of different natures, and each one, as a rule, has to bear a repetition of its own particular load constantly and continuously. The nature of the load should determine the magnitude of the greatest stress which the part subject to it should sustain. If it be a steady one all materials will withstand much higher stresses than are possible with a load more or less suddenly applied and removed continuously. Experience has shown that in structures on land metal can be loaded so that the stress is considerably more than half its elastic limit, if it is steady—that is, if there is no variation or infrequent application; on the other hand, it has been found that if a load of such magnitude is applied and removed quickly and continuously the metal soon gives way. If the stress is lowered either by reducing the load or increasing the section of material to sustain it, the number of applications of load to produce fracture is larger; if further lowered, it takes a larger number still; finally, it is found that if the load does not stress the material beyond a certain fraction of the elastic limit, the number of applications may be so large without producing fracture that the load may be considered quite a safe one, and the corresponding stress used for calculations when employing that material under similar conditions Such loads as these may, for convenience, be called *intermittent*, and the stresses produced *intermittent stresses*.

The parts of the engine subject to intermittent stresses are the piston-rod ends in and beyond the pistons and cross-heads, the piston-rod and connecting-rod caps and bolts, the cylinder cover studs, main bearing caps and bolts, the bolts and studs of the valve gear, the guides, etc., the crank-shaft, the high-pressure cylinder and its covers, the medium-pressure cylinder and its covers generally, and all the column and framing bolts. The tunnel or intermediate shafting theoretically has a *steady load*, but really it is moderately intermittent —that is to say, the variation is not from zero to the maximum, but from a minimum considerably above zero to the maximum.

If the load varies from a positive to a negative value—that is, from tension to compression and from compression to tension—the load is said to be an *alternating* one and the stresses on the material *alternating stresses.* Wohler, of Berlin, had shown, and Professor Unwin, in this country, supports the theory, that the effect of a positive and negative load acting alternately continuously is nearly the same in its destructive effect as a positive load of the same magnitude as the two jointly. That is, if a piston-rod is subject to

TABLE LXXXIII.—LIMITS OF SAFE WORKING STRESSES ON VARIOUS METALS (TENSILE CONTINUOUS SERVICE).

Metal or Alloy.	Nature of the Load.	At Revolutions per Minute, not exceeding									
		60	100	200	300	400	500	600	700	800	1,000
Iron castings of good quality,	Steady	5,150	5,000	4,800	4,700	4,600	4,550	4,500	4,450	4,450	4,440
	Intermittent	4,700	4,500	4,200	3,940	3,750	3,600	3,440	3,300	3,200	3,000
	Alternating	3,900	3,700	3,350	3,100	2,900	2,700	2,550	2,400	2,300	2,200
Mild steel castings, good quality,	Steady	15,200	14,650	13,800	13,100	12,550	12,050	11,600	11,200	10,800	10,150
	Intermittent	13,450	12,800	11,950	11,250	10,650	10,150	9,650	9,200	8,850	8,200
	Alternating	11,150	10,550	9,750	8,950	8,400	7,800	7,330	6,900	6,500	6,000
Mild steel bars and plates,	Steady	16,800	16,500	15,900	15,450	15,100	14,800	14,550	14,350	14,150	13,800
	Intermittent	14,350	14,000	13,400	12,900	12,500	12,200	11,930	11,700	11,500	11,150
	Alternating	12,100	11,750	11,100	10,600	10,250	9,900	9,600	9,360	9,150	8,800
High tensile steel bars and forgings, nickel, chrome, vanadium,	Steady	26,800	26,350	25,500	24,800	24 260	23,800	23,350	22,950	22,550	21,750
	Intermittent	24,350	23,850	22,950	22,250	21,700	21,200	20,750	20,300	19,900	19,100
	Alternating	22,100	21,650	20,700	20,000	19,420	18,900	18,450	18,050	17,700	16,950
Bronze, or gunmetal, Admiralty alloy,	Steady	7,800	7,600	7,200	6,930	6,700	6,500	6,350	6,200	6,100	5,950
	Intermittent	7,000	6,800	6,350	6,000	5,660	5 409	5,150	4,930	4,720	4,350
	Alternating	5,900	5,600	5,100	4,650	4,300	4,000	3,750	3,500	3,300	3,000
High tensile bronzes, carefully cast,	Steady	12,700	12,500	12,100	11,700	11,550	11,250	11,050	10,850	10,700	10,400
	Intermittent	11,450	11,000	10,300	9,800	9,350	9,000	8,700	8,400	8,150	7,500
	Alternating	9,500	9,000	8,300	7,750	7,300	6,930	6,600	6,300	6,000	5,550
High tensile bronze bars, etc., unannealed,	Steady	22 500	21,600	20,800	20,100	19,600	19,200	18,900	18,700	18,500	18,140
	Intermittent	19,500	19,000	18,000	17,200	16,600	16,030	15,500	15,000	14,500	13,600
	Alternating	15,500	15,000	14,000	13,200	12,550	12,000	11,500	11,000	11,600	10,750
High tensile bronze bars, etc., annealed,	Steady	15,250	15,000	14,520	14,040	13,860	13,500	13,250	13,020	12,860	12,500
	Intermittent	13,750	13,200	12,360	11,760	11,120	10,800	10,440	10,089	9,800	9,100
	Alternating	11,400	10,900	9,960	9,300	8,760	8,320	7,920	7,560	7,250	6,750

For shearing stresses, multiply the above by 1·62 for cast iron; 0·8 for steel; strong zinc bronzes, 0·60; delta metal, 0·7; phosphor bronze, cast, 1·10; rolled, 0·63.

a thrust of 20 tons and a pull of 15 tons or *vice versâ*, the effect on the material is nearly the same as if it were subject to an intermittent load of 35 tons. It is, however, not quite the same, and further experiment with modern metals may show that at moderate loads the difference between alternating and intermittent is not so great. In any case, material subject to alternating loads must not be stressed so highly for any one direction as if those loads were intermittent; in fact, from Wohler's experiments the stress should not be much more than half. The rapidity of intermission and alternation is also an important factor in determining safe stresses for materials.

Further, the shape of the part subject to load is of importance; if there are incisions similar to that of a screw thread, which suddenly change and concentrate stress and invite fractures the stresses should be much lower than those of a plain uniform section, such as that of a bar could safely bear.

The parts of an engine subject to alternating stresses are the piston-rods, connecting-rods, cross-heads, columns, pistons, low-pressure cylinder, its cover and bottom, the medium-pressure cylinder, etc., when at low powers, paddle shafts and screw shafts at their outer ends due to the weight of propellers and the reaction of floats, or blades, especially in the case of a screw ship when racing or running light. The valve motions, the air-pump levers, rods, etc., are subject to partially alternating stresses.

Table lxxxiii. gives the safe limits to which an engineer may go under the various conditions, if weight is of prime importance in designing engines for continuous work. If weight is not an object of first importance a reduction of 10 per cent. may with advantage be made. If the article is nicked, suddenly changed in section, or has a screw thread, a reduction of 25 per cent. should always be allowed. In the case of alternating stresses the equivalent force one way only is taken, for calculation—viz., tension.

Shearing and Torsion.—Certain metals have a very high power of resistance to tension and compression without the like ability to resist shearing; for example, naval brass has a coefficient of shear of only 55·8 per cent.—that is, its strength against shearing is little more than half what it possesses against tension; on the other hand, cast iron has a coefficient of 162 per cent., and phosphor bronze castings 113 per cent. The coefficient of wrought steel and iron is 80, while that of most of the strong zinc bronzes is, as a rule, only little more than naval brass. They are consequently not very fit for such purposes as shafting, where shearing forces are always of importance, especially so in the case of tunnel shafting.

The Elastic Limit or Yield Point.—It is probable that every material which has never been subject to direct tension will have some degree of permanent set after application of any load; it will, of course, be very small, and generally microscopic. If, however, the material is stressed nearly to what we call the elastic limit, released, and again subjected to stress, there should be no further permanent set till that point (the elastic limit) is exceeded.

When this limit is found there is reason to suppose that so long as *no portion* of the material is subject to a stress in excess of it, the material should endure for ever. Unfortunately materials do give way from what is called "fatigue" without apparently having been subject at any time to a stress nearly approaching the supposed elastic limit. It follows, then, that the safe limit of continuous stress is considerably below the elastic limit, or that the loads on *some portion* of such structures as have collapsed, are much greater than generally supposed. It is highly probable that the latter is the case, and to a great extent the cause of accident. In a complex structure it is nearly impossible to exactly proportion each part to its load, and absolutely impossible to spread the sustension of load evenly over every inch of area of section. Moreover, in the process of manufacture initial stresses are often set up in many places, all of which are of uncertain magnitude and direction; to these are added the stresses produced by working load, which may not be, and probably are not, distributed in accordance with theory. It is, therefore, more than improbable that a structure designed with a theoretical margin or factor of safety of ample magnitude has in one or more places material stressed dangerously near to the elastic limit, and perhaps sometimes even beyond it in some cases.

Safe Working Stresses.—The limits given in Table lxxxiii. may be used when the maximum stresses are calculable and distributed uniformly over the sections. For example, a weight suspended by a bar having an eye at each end, so as to hang freely, will cause a uniform stress on the transverse sections; if instead of eyes forged with the rod the rod was screwed and fitted with nuts whose faces are rough and the surfaces on which the nuts sit are uneven, the stress will not be uniform, and may be much greater on one side than the other. Again, the sections of a round or square bar loaded and subject to bending either as a beam or cantilever are not uniformly stressed, the outer layer, or that furthest from the neutral axis, being much more so than those nearer it; but at the same time, the layer being firmly attached to its neighbour, receives support from it in a different way from what is the case of a fagot of strips free to slip one on the other as in a carriage spring. The rules for estimating are based, however, practically on the latter supposition, hence what would seem to be a too high stress can be safely borne by such parts.

In the case of a complex article like a crank-shaft subject to complicated loads, it is impossible to estimate the maximum intensity of stress at any and every part of a section. It is certain that at very few, if any, places the stress is uniform over the whole section at such places; and at some the intensity of stress must be great on portions of the sections. Moreover, the quality of the material is not uniform throughout such a forging, the strength across the direction of flow of metal being less than given in Table lxxxiii. Under such circumstances the nominal stress adopted for calculations should be much less than those given in that table. For similar parts of light machinery the nominal stress may be taken at 60 per cent. of those given in this table, and for those of ordinary machinery 35 to 40 per cent. In practice, therefore, the designer must carefully consider each case by itself, with all its conditions and surroundings, to determine the nominal stress on which to calculate the dimensions of the parts which must sustain the load.

Stretch under Tension.—Rigidity of structure is often of prime importance, and generally so in the engine-framing, guides, etc. All materials stretch more or less under stress, and the lower the stress the less is the stretch; consequently it is often found necessary to make details of the structure larger than necessary for mere strength, in order to avoid the stretching of them, which would be inconsistent with rigidity or stiffness of form. It is generally forgotten that up to 7 or 8 tons' tension per square inch cast iron stretches more than steel or wrought iron, and that the material has practically no elastic limit, for it has no permanent set up to fracture.

Again, there are other conditions which must not be overlooked, as, for instance, in the case of castings as distinguished from forgings or other wrought forms of material. The latter are practically free from internal defects* when machined and made part of an engine; the castings, however, *may* be solid, but are often not so, and must not be assumed to be. To allow for such contingencies a reduction of 10 to 20 per cent. should be made from the stress given by sound bars in the testing machine as the elastic limit of castings. The figures in Table lxxxiii. for castings are based on this allowance. Again, rough forgings often have incipient flaws arising from the indenting of the hammer in forging, and, in the case of steel, from small gas bubbles and scoria near the surface of the ingot, and sometimes from slag getting forged on the metal. From the bright working parts such imperfections are machined away or are cut out before finish forging is done.

Another serious consideration in determining the nominal stress for calculation is the purpose the engine has to serve. Express steamers, such as passenger and mail boats, have to exert 80 to 90 per cent. of the trial or maximum power for 50 to 70 per cent. of the year; cross-channel steamers (short runs) steam about 25 to 35 per cent. of the year; while excursion and river steamers do from 30 to 50 per cent. They all have occasionally, and are always liable to have, to run at full power. Every part of the machinery of such ships should, therefore, be much lower stressed at full power than the corresponding parts of machinery which is seldom or never worked to the full power for which it was designed, whose full speed on service is done with only 75 per cent. of such power, and that for only 10 per cent. of each year.

The speed of the engine also affects the problem. A slow-working paddle-engine, with few reversals of motion and time for "cushioning" and other means of avoiding shock, does not jar its component parts as does a high-speed one, the momentum of whose moving parts materially influences the stresses on them and their connections, which stresses are most suddenly applied and removed or reversed many more times in a minute. and which may be increased by water in the cylinder to an unknown extent, and from want of time and other causes may not get the cushioning at the right periods to do good and ease the concussions. Such engines, however, are generally required to run at full speed for very short spells and those far apart, and may, therefore, have lighter scantlings than would be necessary for a similar engine on shore, such as an electric light or power engine, which has to run every day for five to nine hours, or even more.

Paddle Engines, when required to be very light, may be designed by referring to the stresses given under 60 revolutions and modifying on the lines laid down above. When weight is not of so much importance as endurance, the stresses may be reduced by 10 per cent. all round.

Naval Engines, Express Steamer Engines, and **Passenger Cargo Steamer Engines** may be designed by taking the figures under 200 revolutions and allowing 10 per cent. reduction all round, and modifying on the lines laid down.

For **High-speed Craft,** such as **Destroyers, Scouts, Third-class Cruisers, etc.,** the column under 300 revolutions, with the modifications, but without the all-round 10 per cent. reduction, may be followed.

* This is not true of large shafts, for the steel near the axis is never as good as that near the surface and sometimes is very poor indeed, hence the boring out is really no practical loss of strength.

For **Tramp Steamers**, whose power is small, so that the engines are virtually always working at full power, and made of material untested and generally of common quality, a reduction of 20 per cent. from column 100 or 200 revolutions is advisable when calculating the various parts of the engines and machinery.

Boilers, etc., form the most important part of the machinery from the weight point of view, and the possible saving is greater than from any other part, as has been seen when water-tube boilers have been substituted for cylindrical, and earlier when steel shell plates took the place of iron ones. The Admiralty have reduced the thickness of these plates till little remains possible in that direction with ordinary steel, and it is understood that the Board of Trade, Lloyd's, and other authorities will shortly follow suit.* It is also equally possible now that a material as superior to the ordinary mild steel as it was to iron will soon be placed on the market for the use of boiler-makers. It is possible, however, to make a much lighter boiler with the ordinary steel without running any risk beyond what already exists. For example, the steel bars for stays are, when the boilers are in use, subject to a steady continuous load, gently applied and released; they are tested mechanically under similar conditions to a stress of 30 tons per square inch; but when at work are stressed only to 4 tons, whereas 6 tons would not be excessive, especially in these days of non-corrosion in the inside of boilers. With a reduction in the size of stays a saving would be made on the nuts, etc. The inner plating is also heavier than it need be under the above conditions. The heads and snaps of rivets are made regardless of weight, and of the fact that most of the rivets are subject to shear only. For cleaning purposes boilers were and are still designed with room in various parts sufficient to admit a man and permit him to do the work. This means a larger boiler and much more water than is necessary. Now, with salt water cut off and practically pure water used, there cannot be the risk of scaling, especially in short voyages, to warrant such extravagance in space. Railway locomotives use hard water and water containing much solid matter, but no such provision is made as in ordinary marine boilers for excavating scale. Of course where there is a strong probability of salt water being used some such provision must be made, but this would be only in ships where weight of machinery is unimportant and long voyages the rule.

Cost.—Although less material is used when the weight of an engine is cut down, the value of that saved is far exceeded, as a rule, by the cost of labour employed in effecting the reduction. The substitution of one material for another with the object of saving weight invariably means a more costly one. The removal of surplus materials from all forgings, and some of that even from castings, is a costly process, although the modern machine tool, with the high-class self-hardening steels now in use, permits of it being done at a much cheaper rate than formerly obtained. Also, it is generally found that the reducing to a minimum of the sizes of flanges, facings, etc., entails much more care in moulding, marking off, and machining, with the consequent greater cost of labour; besides which, more costly methods of doing work are necessitated by the want of room in carrying it out. Then, too, the much higher number of revolutions resorted to with the object of reducing weight per I.H.P. requires much greater care in the manufacture and adjustment of all the working parts, and the use of fittings and appliances which were done without before.

* The water-tube boiler is exclusively used now in naval ships, and is gradually being supplied to express steamers of all kinds where weight is of serious consequence.

In 1880 a compound naval engine cost about £12 per I.H.P., or £75 per ton. Five years later triple engines of large size were £6·4 and small £8 per I.H.P., and still about £75 per ton. In 1890, the cost of cruisers' machinery was about £7 per I.H.P. and £90 per ton. Later on large cruiser machinery with Belleville boilers cost about £12 per I.H.P. and £130 per ton. The more modern with less expensive water-tube boilers cost somewhat less.

The very light naval engines of the past few years illustrate what can be done in the way of saving weight when cost is of secondary consideration. At the same time it will be seen that the cost per I.H.P. of trial at full power is really low, and even if the lower power or that developed without forcing be taken, it is seen that these engines by comparison are by no means costly. For example, the third-class cruiser engines of 1891, with cylindrical boilers, cost £6·75 per I.H.P. full power, at natural draught it was only £10, or £93 per ton. The third-class cruiser of 1897 had water-tube boilers and engines costing £6·5 per I.H.P. of full power, or £9 of that at natural draught; this was, however, £118 per ton. The destroyer engines of 1895 cost only £6 per I.H.P., but £200 per ton, and later ones cost as much as £236 per ton.

The machinery as now fitted into naval ships is much more elaborate than formerly obtained, and turbines are employed to drive the screws. Battleship machinery, including auxiliaries, in 1913 cost about £9·5 per S.H.P., while that of first-class cruisers cost £8·5 to £7, depending largely on the rate of revolution. The machinery of second-class cruisers and large scouts cost about £6·8 to £7 per S.H.P., while that of destroyers was somewhat less, due to the higher rate of revolution. The cost per ton was about £120 for battleships and £145 for cruisers. Destroyers and similar craft had then machinery costing about £220 to £240 per ton.

Taking similar periods for the mercantile marine, the average cost of a compound engine in 1880 was £7·5 per I.H.P., or £35 per ton. Five years later the triple engine was costing £6·5 per I.H.P., or £36 per ton. In 1890, when prices were high in consequence of the Naval Defence Act having filled most of the shipyards and engine works with orders, engines could be got for under £6 per I.H.P., but still £36 per ton. Ten years later the engines of cross-channel steamers, whose weight would be only about 220 lbs. per I.H.P., could be purchased at £5·5 per I.H.P., or £53 per ton, and heavy engines at about £6·5 per I.H.P., and £35 per ton.

Comparing the cross-channel boats' machinery with that of the 1891 third-class cruiser, it will be seen that per I.H.P. they cost the same money, but the cruiser engines weigh only 156 lbs. against the 220 of the merchant ship. The latter cost, therefore, only £53 per ton against the £93 of the Naval ship.

Auxiliaries and Appurtenances.—Forty years ago the marine engine was practically self-contained; it had as a parasite an auxiliary feed donkey pump, and in large ships two such pumps. The extensive use of water ballast later on necessitated another and a larger donkey pump, and feed heaters were found to be good things; centrifugal circulating pumps also came to be generally used in large ships somewhat earlier. In the Navy, in consequence of the high revolutions, it was found better to have independent feed pumps, and, later, to take from the main engine all pumps except the air pump. This practice was soon followed in the mercantile marine with fast running engines. Then fresh water for making up the water from the boilers was deemed necessary, and evaporators or distillers were used. Starting and turning engines were known then, but have been used very generally since; they are now regular fittings in ships of all sizes. Feed-water filters are common where ships are engaged in long voyages and use oil in the cylinders. All these things add to the weight of the machinery without contributing anything directly to the I.H.P. used as divisor. The I.H.P. developed in the auxiliary machinery of a warship is now enormous. In a large merchant steamer, especially long voyage express ones, as already fully set forth in Chap. xx., the auxiliaries add very much to the weight and steam consumption.

Spare Gear also adds to the weight, and is more in extent to-day than was formerly the case in the mercantile marine; but less in the Navy, where in old days nearly half an engine was carried (*v.* Appendix D).

Fuel.—The consumption of fuel is another most important factor in all considerations of weight; especially is this the case where, as in the North Atlantic, the distance between coaling stations is long. It is of little use paring down the weight of the machinery of ships on this service, if by so doing the fuel consumed per I.H.P. is increased. The great cry of the opponents of the water-tube boiler was that the increased consumption of fuel outbalanced the saving in weight, so that, for a given voyage, the total weight of machinery and fuel together was greater with a water-tube boiler installation than with a cylindrical one with Howden's or Ellis' forced draught. Forty years ago 2·5 lbs. per I.H.P. was the general rate of consumption of good coal; a little later it was as low as 2 lbs. with the very large three-crank compound engines and high pressure of steam. The triple engine in its early days consumed 1·7 lbs. in a general way, and 1·5 under best conditions. Now the best engines with Howden's system, superheaters, feed-heaters, etc., consume if triple-compound 1·4, if quadruple 1·25, if geared turbines 1·15 lbs. of good mercantile coal per hour.

In the Navy the consumption of coal (including auxiliaries) was with triple-compound engines and the cylindrical boiler 1·8 lbs. of Welsh coal when running full speed. Now with turbines and water-tube boilers, etc., it is from 1·55 to 1·70 on service. If the I.H.P. of the auxiliaries were included the rate would, of course, be somewhat lower in both cases.

The modern naval engine when reciprocating has a low rate of expansion when working at full power, as already explained, consequently the fuel consumption per I.H.P. compares unfavourably with that of a merchant ship; but the conditions are dissimilar in more ways than pertain to the main engines; if the main engines and their own particular auxiliaries only are using steam from the boilers the consumption of good Welsh coal does not exceed 1·85 lbs. per I.H.P. The consumption of fuel in recent ships having turbines only is about 1·87 lbs. per S.H.P., which would be equivalent to 1·758 lbs. per I.H.P., assuming an efficiency of 94 per cent. In the cruiser "Dartmouth" it was only 1·48 lbs. at full speed, and 1·72 lbs. at 90 per cent. of full power. The water consumption in scouts running at 500 to 600 revolutions per minute was only 14·85 lbs. at full power, and 15·5 lbs. at half power. H.M.S. "Gloucester" consumed 1·48 lbs. of fuel at full power of 19,000 S.H.P., and at 14,000 it was 1·59 lbs.

The Weight of Modern Marine Machinery Installations of all kinds varies with the rate of revolution of the motor, whether it be a reciprocator or a turbine, when all other things are *pari passu*; it may be arrived at with a very fair approach to accuracy by the following empirical formulæ where I.H.P. and S.H.P. are the mean powers developed on trials at full speed for a period exceeding one hour continuous steaming at revolutions R per minute. Weight found in this way will form a sufficient guide for the preliminary design of a ship, but must not be taken as being quite so accurate as to preclude the necessity for a proper estimation. In naval ships and express passenger steamers having light or small tube water-tube boilers of the Yarrow or other similar type, and with the arrangement and supply of auxiliaries usual in such ships.

$$\text{The total weight of machinery with turbines} = \frac{\text{S.H.P.} \times 16{\cdot}85}{R - 100} \text{ tons.}$$

$$\text{The total weight of machinery with reciprocators} = \frac{\text{I.H.P.} \times 7{\cdot}2}{R - 100} \text{ tons.}$$

Naval ships with reciprocating engines and large or heavier type of water-tube boilers, such as the Babcock, Belleville, Niclausse, etc.

$$\text{Total weight of machinery} = \frac{\text{I.H.P.} \times 14{\cdot}5}{R + 50} \text{ tons.}$$

Mercantile ships having cylindrical boilers and the installation of auxiliaries usually found in them and large naval ships with similar boilers.

$$\text{Total weight of machinery} = \frac{\text{I.H.P.} \times Q}{R + 50} \text{ tons.}$$

(1) Mail steamers with reciprocators of heavy design, . . $Q = 20{\cdot}5$
(2) Mail and naval ships with reciprocators of light single-ended boilers, ... $Q = 20{\cdot}0$
(3) Mail and naval ships with reciprocators of light double-ended boilers, ... $Q = 19{\cdot}5$
(4) Mercantile ships having turbines and cylindrical boilers, . $Q = 44{\cdot}0$
(5) Mercantile ships, such as tramp steamers with heavy machinery, ... $Q = 27{\cdot}0$

Example (1).—To find the weight of machinery of a battleship having turbines and water-tube boilers, the S.H.P. is 25,000 at 300 revolutions per minute.

$$\text{Weight} = \frac{25{,}000 \times 16{\cdot}85}{300 - 100} = 2{,}107 \text{ tons.}$$

Example (2).—The weight of a scout's machinery developing 16,000 S.H.P. at 750 revolutions per minute.

$$\text{Weight} = \frac{16{,}000 \times 16{\cdot}85}{750 - 100} = 415 \text{ tons.}$$

Example (3).—An Atlantic steamer has turbine machinery and cylindrical boilers, the S.H.P. is 30.000 at 180 revolutions.

$$\text{Weight} = \frac{30{,}000 \times 44}{180 + 50} = 5{,}739 \text{ tons.}$$

Example (4).—An express steamer having double-ended boilers and engines of 7,000 I.H.P. at 180 revolutions per minute.

$$\text{The weight is} = \frac{7{,}000 \times 19\cdot5}{180 + 50} = 580 \text{ tons.}$$

Example (5).—A tramp steamer of 1,750 I.H.P. at 70 revolutions per minute.

$$\text{The weight of machinery} = \frac{1{,}750 \times 27}{70 + 50} = 394 \text{ tons.}$$

Relation of Weight to Tonnage.—Taking the same period as before, it is interesting to note that in Atlantic mail steamers the machinery was 13 per cent. of the displacement, and the I.H.P. at the rate of 1·1 per ton of gross register. In small passenger steamers it was about 13 per cent.; the I.H.P. 1·35 per gross register ton.

Ten years later the speed of ocean steamers had increased by 20 per cent. The weight of the machinery was still about 13 per cent. of the displacement, but the I.H.P. was 1·8 per gross register ton.

The increase of speed with the smaller steamers was 25 per cent., the weight of machinery 15 to 19 per cent. of the displacement, and the I.H.P. 1·5 to 1·7 per gross register ton.

In 1895 the Atlantic express steamer had a speed of 21 knots obtained with machinery whose weight was 20 per cent. of the displacement, and whose I.H.P. was at the rate of 1·9 per gross register ton. Twin screw cargo and passenger boats with a speed of 15 knots of this period had a weight of machinery of only 7·3 per cent. of displacement and less than 1 I.H.P. per gross register ton, while in similar boats of 13 knots speed the proportions were 5 per cent. and 0·5 I.H.P.

Fast Channel screw steamers of $17\frac{1}{2}$ knots were at the same time on service with machinery weighing 27 per cent. of the displacement, and indicating 3 to 5 H.P. per gross register ton.

The North Atlantic Service is carried on to-day by express steamers driven by turbines at a speed of 25 knots per hour on the New York route, and by others driven by the reciprocating engine at 22·5 to 23·5 knots, and, lastly, by even larger steamers, like the "Olympic," driven by a combination of reciprocators and turbines at 22·5 knots. The Boston and Canadian service employs ships somewhat smaller, having a speed of 19 to 20 knots and driven by turbines, while the older steamers preceding these more modern ones were driven by reciprocators at about the same speeds.

The machinery of the largest and fastest Atlantic expresses of the turbine type requires about 30 per cent. of the displacement to carry it, and will develop a gross power equal to 2·15 S.H.P. per gross register ton, or 1·90 S.H.P. per ton of gross displacement. That of the "Olympic," develops 1·1 S.H.P. per displacement ton, and the weight is 17 per cent. of the displacement.

The largest Atlantic expresses having reciprocating engines require 23 per cent. of the displacement to carry the weight of the machinery and develop about 2·20 I.H.P. per gross register ton, or about 1·50 I.H.P. per ton of gross displacement. The turbine-driven Canadian and other expresses running at somewhat less speeds require only about 16 per cent. of the displacement, and develop only 1·20 S.H.P. per gross register ton, or 0·75 per ton of gross displacement.

The "Lusitania" and "Mauritania" of the Cunard Line were a great advance on any steamship then existing, being 760 feet long, 87·5 feet beam, and having a displacement of 36,440 tons on 32·5 feet mean draught of water; "Olympic" and her sister ship "Britannic," whose dimensions are, length 853 feet, beam 92·75 and 94 feet, and the latter's displacement is 53,000 tons. The machinery of these ships is the combination of two reciprocating triplex engines, having cylinders 54 inches, 84 inches, and two low pressures of 97 inches diameter, with a piston stroke of 75 inches running at about 80 revolutions, with a huge low-pressure turbine driving the middle screw, as shown in figs. 43 and 44, making about 170 revolutions per minute.

Express steamers of still larger size are the "Aquitania," of 52,000 tons displacement, 870 feet long, and 97 feet beam, and the "Vaterland," 55,000 tons, 908 feet long, 100 feet beam, and a speed about 25 knots.

The Mail and Passenger Steamers on service to India, The Cape, and Australia are also much larger than those engaged in these routes only a few years ago, and although their speed is not so high as in the New York service there is a marked increase. They are generally about 10,000 to 12,500 tons gross register, and steam about 18 knots, being driven by twin-screws and quadruple-compound engines. Their length is usually 500 to 550 feet, with a beam about 60 to 64 feet, and draught of water 27 to 28 feet for the Suez Canal, and rather more on the Cape route. Recently some ships have been placed on the Indian route driven by turbines, and the Union Company, of New Zealand, adopt the mixed method of propulsion in their new steamers.

In 1880 a *Naval First-class Cruiser* attained a speed of 18·5 knots with machinery whose weight was 27 per cent. of the displacement. Ten years later the same speed was recorded with triple-compound engines only 15·5 per cent. in weight. In 1910, for a speed of 24 knots, 17·7 per cent. sufficed, while in second-class cruisers it was 15 per cent., and in third-class cruisers of 24 knots with reciprocators and water-tube boilers 16·8 per cent. Now, with turbines and water-tube boilers wholly, the machinery of a *Battleship* is 15 per cent. of her displacement, develops 2·11 S.H.P. per displacement ton, and drives her at 25 knots. *The Battle Cruiser of* 30 *knots* requires 20 per cent. of the displacement, and develops 3 S.H.P. *The* 30-*knot Light Cruiser* with turbines, etc., has machinery whose weight is 44 per cent. of her displacement and S.H.P. is 11 per displacement ton, while the somewhat larger 26-*knot Cruiser* has machinery 22 per cent., giving 5·5 S.H.P. per ton. *The* 30-*knot Destroyer* with reciprocating engines required 46 per cent. of the displacement for machinery which developed 17 I.H.P. per displacement ton. The modern 32- to 34-knot destroyers have turbine machinery, whose weight is only 30 per cent. of the displacement, and gives 18 S.H.P. per ton displacement and about 60 S.H.P. per ton weight of machinery.

The Modern Battleship is much beyond what was considered unwieldy a few years ago, being now 650 feet long, 92 feet beam, 28,000 tons displacement, about 90,000 S.H.P., giving a speed of 25 knots.

The Battle Cruiser is larger still, being 660 feet long, 90 feet beam, and 30,000 tons displacement; her speed is 30 knots by 87,000 S.H.P. Even *larger Light Cruisers* have been recently built over 900 feet long, with a speed of 35 knots. Turbines and water-tube boilers are in all these modern warships.

The Light Cruiser is now quite a big ship, being 420 feet long, 40 feet beam, 3,800 tons displacement, having engines developing more than 40,000 S.H.P., and driving them at over 30 knots.

The Cruisers of large size of 1907 were 520 feet long, 74·5 feet beam, 14,600 tons displacement, having engines of 28,500 I.H.P., triple-expansion, with water-tube boilers, their speed being 23 knots. The I.H.P. per ton of displacement 1·98, and their weight 16·3 per cent. of the displacement.

Scouts and Destroyers are also of much larger size now than obtained a few years ago, and some of them are of exceedingly high speed. H.M.S. "Swift" is a good example of the development that has taken place of late years to attain high speed with the ability to keep the sea in any weather; she is 345 feet long, 34·2 feet beam, and has a displacement of 2,170 tons on a mean draught of 10·5 feet; she has four screws worked by turbines developing over 30,000 S.H.P., which drive her at about 36 knots. H.M.S. "Tartar," an ocean-going destroyer of 860 tons displacement and a length of 270 feet, has attained a mean speed of 35·4 knots with a little over 14,000 S.H.P. driving three screws. Destroyers are now over 300 feet long.

The Cross-Channel Express Steamer remains at about the same speed as obtained before the advent of the turbine with the fastest of their class; but here, too, there has been a very considerable increase in size of ship during recent years; this is especially so in the English Channel routes. In the Irish Channel express routes the new turbine steamers are 351 feet × 41·1 feet × 24·5 feet deep, of 2,500 tons gross register, and attain a speed of 22·5 knots with three screws; the older boats on the Holyhead and Kingston route are quite as large—viz., 360 feet × 41·5 feet × 27·3 feet deep, with a gross tonnage of 2,641, and a speed of 23·5 knots, obtained with twin screws and reciprocating engines. In the Isle of Man service the turbine steamer, "Ben-ma-Chree," is of 3,353 tons displacement, no less than 375 feet long, and 46 feet beam, and has a speed of 23·27 knots on service, whereas the paddle steamer, "Empress Queen" (fig. 8), was 360 feet long, with a displacement of 3,015 tons, and a speed of 21·7 knots. In the Dover-Calais and Folkestone-Boulogne routes the turbine steamers are now of 2,400 tons displacement, 350 feet long, and 42 feet beam, and attain a speed of 23·5 knots against the 1,700 tons, 338 feet and 34·7 feet beam with 21·5 knots of 1900.

In the Clyde Estuary and other similar service the three-screw turbine steamer has been taking the place of the paddle steamer, and fig. 6 is a good instance of this kind of ship. The speed on this service is quite high enough at 20 to 21 knots, but it is very likely to be increased to even 24 on the longer routes requiring the passage in a day.

In Shallow Water of Rivers, Estuaries, and Lakes the paddle-wheel, especially as fitted at the stern, continues to be the favourite for high speed or great power for towing, especially in America, where much of the passenger traffic is still water-borne. On the great lakes there are huge paddle-boats similar to that shown in fig. 5, but the twin-screw ship is entering into competition, and sooner or later the three- and four-screw turbine ship will enter the field. On the big rivers the stern wheeler has a strong hold on the respect of the shipowner. On the Mississippi there are such boats 300 feet long, 42·6 feet beam, and only 7 feet deep, having large passenger accommodation, and a speed of nearly 18 miles per hour. The engines are simple expansive-condensing, having cylinders 30 inches diameter and piston stroke of 120 inches.

The weight of paddle-wheel machinery is greater than that of the screw, but with modern designs 8 to 9 I.H.P. is developed per ton, as against the 6 to 7 I.H.P. of the light engines and low-pressure steam of thirty years ago.

When circumstances permit of the adoption of the special stern arrangements, as shown in figs. 1 and 2, twin screws of small size and high revolutions can be used with great advantage, so far as weight is concerned.

TABLE LXXXIV.—CONDITIONS OBTAINING WITH MARINE MACHINERY OF THE RECIPROCATING TYPE, 1913.

	Express Steamers.			Steamers for Cargo and Passengers.			Cargo Steamers.			Naval Ships.				Paddle Steamers.		
	Atlantic.	General Ocean.	Cross Channel.	Atlantic.	Medium.	Small.	Large.	Medium.	Small.	1st Class.	2nd Class.	3rd Class.	De-stroyers.	Large.	Medium	Small.
Displacement, tons,	30,000	15,000	3,000	30,000	10,000	3,000	30,000	10,000	5,000	15,000	6,000	3,000	360	5,000	1,000	500
Speed on trial, - -	24·0	19·0	21·0	17·0	15·0	13·0	12·0	11·0	10·0	24·0	21·0	20·5	30	22·0	19·0	16·0
Revolutions, - -	80	80	175	85	80	100	70	80	90	140	180	250	390	50·0	52·0	55·0
Piston speed, - feet,	950	800	900	800	700	700	600	600	550	980	900	1,000	1,170	700	600	400
Boiler pressure, - lbs.,	220	200	180	200	200	180	180	170	170	205	300	300	250	140	120	120
Referred mean pressure ,,	36·0	38·0	45·0	39	36	34	33	30	30	45	50	52	47	42·0	40·0	38·0
Referred mean ÷ absolute boiler pressure,	·153	·177	·236	·166	·167	·174	·169	·162	·162	·205	·156	·165	·177	0·300	0·333	0·316
Total weight of machinery per I.H.P., lbs.,	336	330	224	490	438	336	408	400	336	215	190	120	51	300	280	300
Weight of machinery ÷ displacement,	0·23	0·125	0·268	0·096	0·060	0·125	0·042	0·050	0·047	0·200	0·148	0·178	0·460	0·360	0·375	0·400
I.H.P. per ton of machinery weight,	6·75	6·80	10·0	4·6	4·880	6·75	5·50	5·60	6·67	8·36	9·36	15·2	...	...	...	...
I.H.P. per sq. ft. of grate,	13·0	11·50	13·5	12·5	12·5	12·5	11·75	11·50	10·5	10·72	9·84	16·24	...	...	...	...
I.H.P. per 100 sq. ft. of total heating surface,	35·0	35·0	40·0	34·0	36·0	37·0	34	33	33	30·4	32·0	32·8	...	...	...	...
I.H.P. per ton gross register, - -	2·15	1·75	4·0	0·60	0·58	1·25	0·45	0·46	0·65	...	...	...	...	5·00	4·00	3·0
Fuel per I.H.P. per hour, - lbs.,	1·30	1·27	1·60	1·27	1·27	1·45	1·30	1·45	1·5	1·80	1·85	1·87	2·5	2·00	2·10	2·25
I.H.P. force draught per ton, - -	6·9	7·0	10·5	4·73	5·10	8·0	...	...	...	10·4	11·7	19·0	43·6	7·47	8·00	7·47
I.H.P. force draught per sq. ft. of grate,	18·0	18·0	20·0	16·0	16·0	18·0	...	...	...	13·4	12·3	20·3	26·0	15·0	13·0	10·0
I.H.P. force draught per 100 sq. ft. of total heating surface,	45·0	45·0	52·0	36·0	38·0	40·5	...	...	...	38·0	40	41·0	45·0	45·0	42·0	40·0
Ratio L.-P. to H.-P. cylinder, triple,	7·5	7·0	6·0	8·0	8·0	7·0	7·5	7·2	7·0	6·3	6·8	7·1	5·75	5·0	6·0	...
Do. do., quadruple,	9·0	8·3	...	8·8	8·7	8·0	8·9	8·3	8·0	...	...	...	...	...	...	...
Do. do., compound,	...	...	...	...	...	...	...	...	...	...	...	...	...	3·5-3·85	3·7-4·0	3·5-4·0

TABLE LXXXV.—Particulars of Naval Ships' Machinery

Designation.	When Built.	Number of Screws.	Kind of Engines.	Cylinders. Number.	Cylinders. Diameter.	Cylinders. Stroke.	Full Speed. Revolutions per Minute.	Full Speed. Piston Speed.	Full Speed. Referred Mean Pressure.
					Ins.	Ins.		Ft.	Lbs.
Battleship	1870	1	Horizontal Trunk Surface-Condensing, -	2	127 (= 118)	54	71·5	643	20·1
Do.,	1870	1	Do. Return Connecting Rod Surface Condensing, - - - -	2	120	54	62·7	564	19·5
Do.,	1870	1	Do. do. do. do.,	2	98	48	68·4	547	19·6
Do.,	1870	2	Do. do. do. do.,	4	72	36	72·2	433	23·0
Do.,	1880	2	Vertical Direct-Acting Compound, - -	4	60-104	42	82·1	575	22·3
Do.,	1885	2	Do. do. do., - -	6	77-55-77	48	80·0	640	25·8
Do.,	1888	2	Do. do. do., - -	6	..	45	100·0	750	..
Do.,	1890	2	Do. do. Triple, - - -	6	43-62-96	51	95·0	807	34·8
Do.,	1895	2	Do. do. do., - - -	6	40-59-88	51	103·0	875	33·1
Do.,	1897	2	Do. do. do., - - -	6	40-59-88	51	103·0	875	38·4
Do.,	1899	2	Do. do. do., - - -	6	30-49-80	51	108·5	922	49·0
Do.,	1901	2	Do. do. do., - - -	6	31½-51½-84	51	110·2	936	50·5
Do.,	1903	2	Do. do. do., - - -	8	33½-54½-63-63	48	120·0	960	50·2
Cruisers— 1st Class,	1870	1	Horizontal Trunk Surface-Condensing, -	2	(= 104¼)	48	74·1	593	24·0
Do.,	1877	2	Do. Return Connecting-Rod Compound,	8	41/75-41/75	36	97·2	553	24·8
Do.,	1885	2	Do. do. do. do.,	4	46-86	39	95·1	618	26·1
Do.,	1888	2	Do. Direct-Acting Triple, - - -	6	36-51-78	42	116·0	812	37·1
Do.,	1890	2	Vertical do. do., - - -	12	36-52-80	48	105·0	840	41·8
Do.,	1894	2	Do. do. do., - - -	6	40-59-88	51	106·0	900	33·4
Do.,	1897	2	Do. do. do., - - -	8	45-70-76-76	48	113·0	904	51·7
Do.,	1899	2	Do. do. do., - - -	8	34-55½-64-64	48	116·6	932	46·7
Do.,	1901	2	Do. do. do., - - -	8	34-55½-64-64	48	118·9	951	50·3
Do.,	1902	2	Do. do. do., - - -	8	36-59-68-68	48	120·0	960	50·6
Do.,	1902	2	Do. do. do., - - -	8	43½-71-81½-81½	48	122·4	979	50·8
Do.,	1906	2	Do. do. do., - - -	8	43½-69-77-77	42	135·0	935	44·1
Do.,	1908	2	Do. do. do., - -	8	40½-65½-75-75	48	127·0	1016	50·7

TABLE LXXXV*a*.—Particulars of Naval Ships' Machinery

Designation.	When Built.	Number of Screws.	Kind of Engines.	Cylinders. Number.	Cylinders. Diameter.	Cylinders. Stroke.	Full Speed. Revolutions per Minute.	Full Speed. Piston Speed.	Full Speed. Referred Mean Pressure.
Cruisers— 2nd Class,	1870	1	Horizontal Trunk Surface-Condensing, -	2	(=86)	45	79·1	593	21·5
Do.,	1870	1	Do. Direct-Acting Surface-Condensing,	2	88	42	74·2	520	20·3
Do.,	1877	1	Do Return Connecting-Rod Compound,	3	88-72-88	48	70·0	560	24·0
Do.,	1882	2	Do. Direct-Acting Compound, - -	4	42-80	45	90·0	675	26·7
Do.,	1886	2	Do. do. do., - -	4	38-64	39	120·0	780	40·4
Do.,	1889	2	Do. do. Triple, - - -	6	34½-51-76½	36	140·0	820	41·6
Do.,	1889	2	Vertical do. do., - - -	6	33½-49-74	39	140·0	910	40·0
Do.,	1891	2	Do. do. do., - - -	6	33½-49 74	39	140·0	910	38·8
Do.,	1896	2	Do. do. do., - - -	6	33-49-74	39	144·2	936	40·3
Do.,	1898	2	Do. do. do., - - -	6	26-42-68	39	141·0	916	50·9
Do.,	1900	2	Do. do, do., - - -	8	26-42-48-48	30	181·3	906	49·2
Cruisers— 3rd Class,	1870	1	Horizontal Return Connecting-Rod Surface Condensing, - - - -	3	55	30	90·3	451	20·0
Do.,	1875	1	Do. Return Connecting-Rod Compound,	2	57-90	33	95·8	527	21·1
Do.,	1878	1	Do. do. do do.,	4	46-64	36	96·0	576	22·1
Do.,	1884	1	Do. do. do. do.,	4	42-72	36	80·3	481	26·3
Do.,	1885	2	Do. Direct-Acting Compound, - -	4	26-46	30	150·0	750	44·4
Do.,	1887	2	Do. do. do., - -	4	27-50	33	150·0	825	33·2
Do.,	1888	2	Do. do. Triple, - - -	6	26-37-57	33	150·0	825	36·4
Do.,	1890	2	Vertical do. do., - - -	6	26½-39-57	27	220·0	990	30·0
Do.,	1890	2	Do. do. do., - - -	6	30½-45-68	33	160·0	880	24·0
Do.,	1891	2	Do. do. do., - - -	6	30½-45-68	33	160·0	880	37·1
Do.,	1898	2	Do. do. do., - - -	6	20½-33-54	27	218 0	981	52·7
Do.,	1901	2	Do. do. do., - - -	8	20½-33 -38½-38½	24	250·0	1000	52·0
Do.,	1903	2	Do. do. do., - - -	8	24¼-38½-42¼-42¼	24	250·0	1000	57·3
Do.,	1904	2	Do. do. do., - - -	8	31½-51½-57-57	30	211·0	1055	44·0

(Battleships and 1st Class Cruisers)—Reciprocating Engines.

Boilers.				Full Power Trial.		4/5 or Natural Draught.		Weight of Machinery.			I.H.P. Full Speed per			Weight ÷ Displacement.
Number and Kind.	Pressure.	Grate.	Total Heating Surface.	I.H.P.	Speed.	I.H.P.	Speed.	Total.	Full Power per I.H.P.	Natural Draught per I.H.P.	Ton of Machinery.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
	Lbs.	Sq. ft.	Sq. ft.		Knots.		Knots.	Tons.	Lbs.	Lbs.				
Box, - - - -	30·0	840	22,200	8,539	14·69	7,187	13·8	1090	286	339	7·83	10·2	38·5	0·126
Do., - - - -	31·6	..	..	7,842	14·97	7,470	14·7	1043	300	313	7·48	..	..	0·129
Do., - - - -	30·0	570	14,388	4,960	13·75	..	..	714	327	..	6·86	8·6	34·0	0·103
Do., - - - -	31·7	515	14,115	4,914	14·0	4,311	13·2	750	342	389	6·55	9·5	33·9	0·125
Cylindrical, - -	65	608	17,500	6,624	14·4	..	..	998	337	..	6·64	10·9	37·9	0·131
Oval and cylindrical,	90	667	22,784	10,184	16·75	8,000	16·2	1266	278	354	8·04	15·3	44·7	0·151
S.E. cylindrical, -	90	790	20,587	11,271	16·75	8,277	16·0	1209	241	331	9·30	14·2	54·7	0·114
Cylindrical, - -	135	629	18,930	12,465	16·75	8,813	16·2	1015	189	258	12·20	19·7	65·8	0·015
S.E. cylindrical, -	155	731	20,348	11,500	17·5	9,430	16·75	1163	224	276	10·00	16·0	56·6	0·082
8 do. do., -	155	817	25,233	12,414	18·0	10,404	17·0	1341	241	282	9·30	15·2	49·3	0·090
Belleville, - -	300	1050	33,770	13,763	18·5	10,454	17·2	1290	209	280	10·70	13·1	40·8	0 100
Do., - -	300	1170	37,120	15,503	18·5	11,626	..	1400	202	270	11·10	13·2	41·8	0·093
Mixed, - - -	..	1390	43,260	18,220	19·0	13,670	..	1580	196	260	11·40	13·1	42·1	0·113
Box, - - - -	30	900	24,868	7,361	16·5	..	..	1043	317	..	7·06	8·18	29·6	0·180
S.E. oval, - - -	65	690	19,500	7,714	18·5	6,658	17·0	1012	294	340	7·62	11·18	39·5	0·271
8 S.E. cylindrical, -	90	546	15,160	5,661	16·6	..	..	776	307	..	7·29	10·37	37·2	0·180
4 D.E. cylindrical, -	130	492	15,190	8,738	18·75	6,090	18·0	772	198	279	11·30	17·7	57·5	0·138
D.E. cylindrical, -	155	1135	31,073	21,411	21·6	14,924	20·4	1543	162	231	13·80	18·8	68·5	0·171
4 D.E. 1 S.E. cylindrical,	155	812	24,908	12,851	20·5	10,517	19·9	1161	202	247	11·10	15·2	51·4	0·158
Belleville, - -	300	2200	67,800	25,774	22·0	22,547	21·5	2230	195	224	11·50	11·7	38·0	0·157
Do., - -	300	1460	40,990	16,961	20·5	12,785	19·8	1540	204	270	11·00	11·6	42·6	0·140
Do., - -	300	1390	47,300	19,156	21·0	14,049	19·9	1577	185	250	12·10	13·8	40·5	0·143
Do., - -	300	1650	51,600	21,400	21·75	16,440	..	1800	188	245	11·90	13·0	41·3	0·150
Do., - -	300	2314	71,964	31,409	24·0	23,103	..	2500	178	242	12·60	13·6	43 6	0·178
Do., - -	300	1610	50,300	22,360	22·7	16,100	..	1750	175	243	12·80	13·8	44·4	..
Mixed, - - -	205	1760	62,250	23,500	23·6	..	..	2250	215	..	10·44	13·3	37·7	..
Do., - - -	210	..	72,000	27,000	23·5	19,000	21·3	..	..	..	..	..	..	..

(2nd and 3rd Class Cruisers)—Reciprocating Engines.

Box, - - - -	30	504	12,748	4,500	15·12	..	..	611	304	..	7·36	9·00	35·4	0·200
Do., - - - -	30	455	12,420	3,878	14·82	..	..	640	370	..	6·06	8·52	31·3	0·209
S.E. cylindrical, -	70	510	12,700	4,950	14·5	..	..	873	395	..	5·67	9·70	38·9	0·252
8 do. do., -	90	542	14,924	5,500	16·6	4,100	..	824	336	448	6·67	10 16	37·0	0·219
Gunboat, - - -	110	388	11,565	6,151	18·0	4,217	..	559	204	300	11·0	15·8	53·0	0·157
4 D.E. cylindrical, -	155	570	13,830	9,653	19·0	6,215	18·0	649	151	234	14·8	16·9	70·0	0·220
Do. do., -	155	525	12,682	9,435	19·0	6,144	18·0	624	148	228	15·3	17·9	74·4	0·221
Do. do., -	150	575	15,641	9,271	20·5	7,423	19·3	748	181	226	12·4	16·1	59·4	0·220
8 S.E. do., -	155	624	18,579	9,846	20·3	8,307	19·6	915	208	247	10·8	15·8	53·0	0 210
Belleville, - -	309	869	25,606	10,272	20·0	7,155	18·7	825	180	258	12·8	11·9	40·1	0·142
Do., - -	300	792	24,426	9,770	20·5	7,660	19·3	832	190	243	11·7	12·3	40·0	0·149
Box, - - - -	30	210	5,980	2,170	12·8	..	..	236	244	..	9·19	10·3	36·3	0·152
6 S.E. cylindrical, -	60	244	5,850	2,139	13·0	..	..	350	374	..	6·01	8·9	36·6	0·165
Gunboat, - - -	60	246	6,312	2,400	13·0	..	..	394	365	..	..	..	38·1	0·165
S.E. cylindrical, -	90	301	9,254	4,030	14·6	3,131	..	517	300	370	7·8	..	..	..
Gunboat, - - -	120	213	6,400	3,365	16·5	2,201	..	296	197	301	11·3	15·8	52·6	0·207
Do., - - -	130	222	6,836	3,754	16·5	2,462	..	353	210	321	10·6	16·9	55·1	0·216
Do., - - -	140	244	7,878	4,613	17·5	2,647	..	395	192	334	11·6	18·9	58·4	0·242
Locomotive, - -	155	244	7,088	4,561	18·5	3,592	17·8	274	135	170	16·7	18·7	64·2	0·149
4 D.E. cylindrical, -	155	365	9,938	..	..	4,615	17·2	515	..	249	..	..	46·5	0·200
Do. do., -	155	410	11,025	7,469	19·5	5,016	17·6	539	156	257	13·8	18·2	69·7	0·209
Express water-tube,	300	352	18,100	7,152	20·7	5,388	19·5	383	120	159	18·9	20·3	39·7	0·179
Do. do.,	300	352	17,760	7,331	20·8	5,218	19·5	370	113	159	19·8	20·8	41·7	0·168
Do. do.,	300	490	26,000	9,800	..	..	..	..	..	..	..	..	37·7	..
Do. do.,	..	752	43,000	14,330	25·2	..	..	..	..	..	..	..	33·3	..

TABLE LXXXV*b*.—Particulars of Machinery of

Designation.	When Built.	Number of Screws.	Kind of Engines.	Cylinders. Number.	Cylinders. Diameter.	Cylinders. Stroke.	Full Speed. Revolutions per Minute.	Full Speed. Piston Speed.	Full Speed. Referred Mean Pressure.
					Ins.	Ins		Ft.	Lbs.
Gunboat,	1876	1	Horizontal Return Connecting-Rod Compound,	2	31-48	18	137	411	23·8
Do.,	1889	1	Horizontal Direct-Acting Triple,	3	20-30-45	24	184	736	36·6
Torpedo gun-boat,	1886	2	Vertical do. do.,	6	22-33-49	18	310	930	..
Do.,	1890	2	Do. do. do.,	6	22-33-49	21	250	875	38·4
Do.,	1890	2	Do. do. do.,	6	24-34-51	21	250	875	34·2
Vidette boat,	..	1	Do. do. Compound,	2	8½-13	9	460	690	54·8
Torpedo boat,	1877	1	Do. do. do.,	2	9½-15	9·5	435	689	47·6
Do.,	1877	1	Do. do. do.,	2	12¾-21	12	354	708	46·5
Do.,	1886	1	Do. do. do.,	2	18-30	16	402	1072	44·3
Do.,	1887	1	Do. do. Triple,	3	15-22-34	16	402	1072	36·4
Do.,	1894	1	Do. do. Quadruple,	4	14-20-27-36	16	410	1093	47·2
Do.,	1893	1	Do. do. Triple,	3	18-26-39½	18	370	1110	55·8
Destroyer,	1893	2	Do. do. do.,	6	18-26-39½	18	360	1078	43·8
Do.,	1894	2	Do. do. do.,	6	18-26-39½	18	393	1178	44·5
Do.,	1894	2	Do. do. do.,	8	19-27-27-27	16	392	1046	60·0
Do.,	1894	2	Do. do. do.,	6	19-29-43	18	366	1098	46·4
Do.,	1895	2	Do. do. do.,	6	18-27½-42	18	366	1097	42·2
Do.,	1895	2	Do. do. do.,	6	18-27-42	18	397	1190	43·7
Do.,	1897	2	Do. do. do.,	8	20-29-30-30	18	395	1186	56·9
Do.,	do.,	2	Do. do. do.,	6	21-32½-48	18	365	1095	53·2
Do.,	do.,	2	Do. do. do.,	6	19-29½-46	18	395	1185	52·1
Do.,	do.,	2	Do. do. do.,	8	20¾-31-34-34	18	392	1176	50·0
Do.,	1904	2	Do. do. do.,	8	23-35-38-38	20	342	1140	..
Gunboat,	1898	2	Do. do. do.,	6	11-17-27	16	286	763	55·3
Sloop,	1901	2	Do. do. do.,	6	11½-18-29½	24	200	800	42·6

Small Naval Ships (Reciprocating Engines).

Boilers.				Full Power Trial.		4/5 or Natural Draught.		Weight of Machinery.			I.H.P. Full Speed per			Weight ÷ Displacement.
Number and Kind.	Pressure.	Grate.	Total Heating Surface.	I.H.P.	Speed.	I.H.P.	Speed.	Total.	Full Power per I.H.P.	Natural Draught per I.H.P.	Ton of Machinery.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
	Lbs.	Sq. ft.	Sq. ft.		Knots.		Knots.	Tons.	Lbs.	Lbs.				
2 Gunboat, - -	60	45	1,018	519	10·0	..	..	79·7	344	..	6·51	11·53	50·9	0·233
Do., - -	150	68	2,200	1298	14·1	1033	12·4	108·3	185	235	12·0	19·10	59·0	0·135
4 Locomotive, - -	150	121	4,426	2696	18·5	..	..	120·0	97	..	23·1	22·4	61·3	0·219
Do., - -	150	190	5,330	3840	20·0	2598	..	171·0	100	149	22·4	20·2	72·0	0·232
Do., - -	150	171	6,204	3665	19·0	2631	..	211·0	130	178	17·3	21·4	59·1	0·287
1 Locomotive, - -	146	5·5	235	151	15·0	..	..	4·6	68	..	33·0	27 5	64·3	0·514
Do., - -	130	7	278	175	15·2	..	..	5·51	70·4	..	31·8	25·0	63·0	0·441
Do., - -	120	15·5	630	400	17·5	..	..	11·0	61·5	..	36·4	26 0	63·5	0·410
Do., - -	130	41	1,775	1013	19·0	..	..	31·6	70	..	32·0	24·7	57·0	0·500
Do., - -	180	38	1,660	1057	19·5	..	..	31·5	66·9	..	33·5	27·8	63·7	0·500
2 Water-tube Yarrow,	250	50	2,350	1600	23·0	..	..	..	..	..	..	32·0	68·1	..
Do. do.,	220	75	3,500	2300	23·5	..	..	48·8	47·5	..	47·1	30·7	66·0	0·377
2 Locomotive, - -	180	100	4,656	3497	26·1	..	..	94·0	60	..	37·2	35·0	75·0	0·392
Water-tube Yarrow,	180	154	8,216	3884	27·6	..	..	90·6	52	..	42·8	25·1	47·3	6·377
Do. Thornycroft,	210	189	9,780	4368	28·4	..	..	108·2	56	..	40·3	23·1	44·7	0·438
Do. Normand, -	175	150	8,800	4484	27·3	..	..	123·2	62	..	36·3	29·9	50·9	0·440
Do. Reed, - -	210	209	10,164	3885	27·9	..	..	116·2	67	..	33·4	18·5	38·7	0·461
Do. Blechynden,	200	176	10,022	4367	27·4	..	..	106·0	54	..	41·1	24·8	43·6	0·400
Do. Thornycroft,	220	196	11,855	5795	30·1	..	..	127·0	49	..	45·6	29·5	48·9	0·462
Do. Normand, -	220	210	11,945	6347	30·1	..	..	143·5	50	..	44·2	30·2	53·0	0·478
Do. Reed, - -	250	255	13,384	6318	30·3	..	..	143·7	51	..	43·9	24·7	47·2	0·479
Do. Thornycroft,	250	245	13,561	6523	30·2	..	..	126·0	43	..	51·7	27·5	48·0	0·420
Do. Yarrow, -	240	270	16,000	7766	26·3	..	..	173·0	50	..	44·8	28·7	48·5	0·346
4 Yarrow, - -	250	80·5	4,000	1300	13·5	960	13·0	95·0	164	..	13·7	16·1	35·5	0·136
4 Babcock, - -	210	144	4,040	1404	13·5	1021	..	162·5	260	356	8·6	10 0	35·0	0·151

TABLE LXXXVI.—Examples of Ocean Express

Name of Ship.	When Built.	Tonnage. Gross Register	Tonnage. Displacement.	No. of Screws.	Cylinders. No.	Cylinders. Diameters.	Cylinders. Stroke.	Revolutions per Min.	Speed of Piston per Min.	Referred Mean Pressure.
		Tons.	Tons.			Inches.	Ins.		Feet.	Lbs.
Oregon,	1883	7,353	13,700	1	3	104-70-104	72	65·3	784	33·0
Umbria,	1884	8,128	15,500	1	3	105-71-105	72	67·5	810	33·6
Oroya,	1886	6,297	10,500	1	3	40-66-100	72	64·5	774	36·6
Victoria,	1887	6,527	8,124	1	3	40·5-63·5-102	72	63·0	756	33·9
Teutonic,	1889	9,686	18,000	2	6	43-68-110	60	78·0	780	37·8
Oruba,	1889	5,857	9,700	1	3	30-66-97	66	67·8	746	33·5
Furst Bismark,	1890	8,430	15,300	2	6	43·5-67-106·3	63	81·0	850	35·1
Empress of India,	1891	5,905	11,500	2	6	32-54-82	54	88·0	792	39·9
Ophir,	1891	6,900	10,600	2	6	34-51·5-85	54	87·5	768	31·2
Scot,	1891	7,815	14,500	2	6	34·5-57·5-92	60	80·0	800	36·0
Campania,	1893	12,950	22,000	2	10	37/98-79-37/98	69	80·9	930	33·0
K. Wilhelm, D.G.	1897	14,349	23,500	2	8	52-89·7-96·4-96·4	69	77·2	887	38·5
Oceanic.	1899	17,274	25,900	2	8	47·5-79-93-93	72	79·0	948	35·8
Savoia.	1900	11,638	15,400	2	8	44·5-68·5-80·3-80·3	67	92·0	1,027	36·5
Deutschland,	1900	16,500	24,500	2	12	104-36·6/106-36·6/106-74	73	80·0	973	37·4
K. Wilhelm II.	1902	19,361	26,500	2	16	37/49-75-112	71	80·0	947	35·0
Caronia,	1905	19,687	27,500	2	8	39-54-77-110	66	75·0	825	46·3
Empress of Britain,	1906	14,190	18,196	2	8	36-52-75-108	69	72·0	828	40·5
K. Princess Cecilie,	1908	19,400	27,000	2	8	37-49-75-113	71	82·0	970	40·8
Laurentic,	1908	14,892	19,160	3	8	30-46-53-53 and 1 L.P. Turb.	54	85·0	765	42·4
Olympic,	1910	46,359	49,336	3	8	54-84-97-97 and 1 L.P. Turb.	75	78·3	978	42·5
Lutetia,	1913	14,580	15,600	4	8	41-57-65-65 and 2 L.P. Turbs.	44	..	..	..
Bergensfiord,	1913	10,666	16,177	2	8	26-37·5-53-75	51	95·0	808	39·3
R. V. Eugenie,	1913	9,727	10,180	4	8	29-43-47-47 and 2 L.P. Turbs.	42	113	791	..
Kaisar-i-Hind,	1915	11,430	15,706	2	8	30·5-44-61-87	54	87·5	788	40·3
Britannic,	1915	47,500	53,000	3	8	54-84-97-97 and 1 L.P. Turb	75	77·0	962	40·6
Justicia,	1917	32,234	41,500*	3	8	25·5-36 64-64 and 1 L.P. Turb.	60	85·0*	850	..

* Approximate figures

Steamers (Reciprocating or Mixed Engines).

Boilers.				Trial Results.		Machinery Weight.		Trial I.H.P. per				Weight of Machinery per Ton of Displacement.
Number and Kind.	Steam Pressure.	Grate Area.	Total Heating Surface.	Total I.H.P.	Mean Speed.	Total.	Per I.H.P.	Ton of Machinery	G.R.T.	Sq. Ft. of Grate.	100 Sq Ft. of T.H.S.	
	Lbs.	Sq. Ft.	Sq. Ft.		Knots.	Tons.	Lbs.					
Cylindrical, . . .	110	1,575	38,062	13,300	18·30	2,205	371	6·03	1·80	8·44	35·0	·161
,, . . .	110	1,680	38,817	14,300	19·00	2,330	365	6·13	1·76	8·51	39·4	·150
6 D.E. Cylindrical, .	160	626	17,640	6,751	16·50	1,343	446	5·03	1·07	10·80	38·4	·128
,, ,, .	150	632	15,610	6,347	17·00	..	..	..	0·97	10·00	40·7	..
,, ,, F.D.	180	1,154	40,972	17,000*	21·00	2,778	366	6·12	1·[illegible]6	14·70	41·5	·103
,, ,, .	160	530	15,107	5,525	16·50	1,257	510	4·40	0·94	10·40	36·6	·130
,, ,, .	155	1,450	47,000	16,000	20·50	2,430	340	6·59	1·66	11·00	34·0	·159
4 D.E. ,, F.D.	160	520	20,193	10,125	18·00	1,414	313	7·16	1·71	19·40	49·9	·123
5 D.E. & 2 S.E. ,, .	160	756	26,004	8,386	18·50	1,290	342	6·55	1·21	11·09	32·2	·122
6 D.E. ,, F.D.	160	837	22,964	11,656	18·50	1,720	330	6·78	1·50	13·90	50·7	·112
12 D.E. ,, .	165	2,630	82,200	29,600	22·00	4,440	336	6·69	2·28	11·20	36·0	·202
12 D.E & 2 S.E. ,, .	178	2,618	84,285	30,000	22·80	4,460	333	6·73	2·09	11·50	35·6	·190
15 D.E. ,, F.D.	192	1,962	74,686	28,000	21·00	4,414	353	6·34	1·54	13·60	35·5	·170
16 S.E. ,, F.D.	175	1,225	45,565	23,000	22·60	2,505	244	9·18	2·09	18·80	50·4	·163
12 D.E & 4 S.E. ,, F.D	220	2,188	85,468	38,900	23·50	5,670	326	6·86	2·36	17·90	45·6	·231
12 D.E & 7 S.E. ,, .	225	3,121	10,643	40,000	23·50	5,780	324	6·91	2·00	12·80	37·2	·218
D.E. ,, F.D.	215	1,298	52,140	22,000	18·50	4,145	422	5·00	1·12	16·90	42·2	·160
6 D.E. & 3 S.E. ,, F.D.	220	1,125	47,141	18,508	19·65	2,383	288	7·77	1·30	16·50	39·3	·131
7 D.E. ,, F.D.	220	2,970	101,900	45,000	23·50	6,530	325	6·89	2·32	15·2	44·1	·242
6 D.E. ,, F.D.	210	777	29,300	13,950	18·00	2,330	380	6·00	0·93	18·00	47·3	·122
24 D.E. & 5 S.E. ,, F.D.	210	5,400	151,000	54,836	22·50	8,580	355	6·30	1·19	10·15	36·3	·163
18 S.E. ,, F.D.	200	1,205	46,950	19,000	20·50	..	..	..	1·30	15·70	40·5	..
8 S.E. ,, F.D.	220	590	23,000	8,500	17·50	1,784*	465	4·82	0·797	14·40	37·0	·176
7 S.E. ,, F.D.	180	480	20,660	10,840	18·10	1,743*	360	6·22	1·11	22·60	52·4	·142
4 D.E. & 4 S.E ,, F.D.	215	743	30,570	11,530	18·00	2,296	446	5·02	1·01	15·50	37·7	·146
24 D.E. & 5 S.E. ,, F.D.	210	5,400	150,958	54,800	22·5	8,600	352	6·37	1·15	10·15	36·3	·162
12 D.E. ,, F.D.	215	1,416	61,344	25,000*	18·0*	4,167*	380	6·00	0·775	17·60	40·8	·104

TABLE LXXXVII.—EXAMPLES OF PASSENGER-

Name of Ship.	When Built.	Tonnage.		No. of Screws.	Cylinders.			Revolutions per Min.	Speed of Piston.	Referred Mean Pressure.
		Gross Register	Displacement.		No.	Diameters.	Stroke.			
		Tons.	Tons.			Inches.	Ins.		Feet	Lbs.
Martello,	1884	3,721	7,800	1	3	31-50-82	57	59·0	561	26·2
Cufic,	1888	4,827	10,600	1	3	27-44·5-74	60	67·0	670	33·3
Cazengo,	1889	2,889	5,180	1	3	32-48-80	48	73·0	584	34·2
Bovic,	1892	6,583	14,500	2	6	22·5-36·5-60	48	80·0	640	34·5
Kensington,	1894	8,669	17,850	2	8	25·5-37·5-52·5-74	54	87·0	783	40·0
Georgic,	1895	10,077	20,000	2	6	24-39·5-65·5	51	77·0	655	34·5
K. Wilhelmina,	1896	4,248	6,575	1	4	27-39-56-80	54	68·2	614	36·0
Othello,	1896	5,059	11,200	1	3	24·75-40-70	48	71·0	568	36·7
Cymric,	1898	12,647	25,000	2	8	25·5-36·5-53-75·5	54	81·0	729	36·9
K. Wilhelm I.	1898	4,446	7,845	1	4	27·5-39-55-82	60	73·0	730	32·6
Sylvania,	1898	5,600	12,160	2	6	22·5-36·5-60	48	94·0	752	41·5
Cleopatra	1899	6,849	13,900	1	3	32-54-90	66	65·0	715	34·4
Saxonia,	1900	13,963	22,580	2	8	29-41·5-59-84	54	78·0	702	39·0
Runic.	1900	12,482	24,600	2	8	22-31·5-46-67	51	81·0	689	35·3
Celtic,	1901	20,880	30,000	2	8	33-47·5-68·5-98	63	80·0	840	35·2
Missourian,	1902	7,914	17,200	2	6	25-42·5-72	48	70·0	560	35·0
Tahiti,	1904	7,585	12,500*	2	6	30-50-80	54	85·0	765	42·1
Amsterdam,	1906	16,967	25,550*	2	8	29-41·5-61-87	60	85·0	850	40·8
Orcomo,	1905	11,546	18,500*	2	8	26-37·5-53·5-76	54	80·0	720	40·9
Rotterdam,	1908	24,149	30,500*	2	8	33-47-68-97·5	60	85·0	850	42·0
Edinburgh Castle,	1910	13,326	20,000*	2	8	32-46-66-96	60	85·0	850	40·2
Franconia,	1911	18,154	24,290	2	8	33-47-67-95	60	77·0	770	37·4
Port Lincoln,	1913	7,243	12,266	1	4	27·5-39-56-81	54	76·8	691	38·0
Orion,	1914	..	19,600	2	6	27-46-76	48	95·0	760	33·2
Insulinde,	1914	9,615	14,080	2	6	29-47-81	52	80·0	693	32·4
Berrima,	1915	11,120	19,789	2	8	23·5-34·5-48·5-70	54	78·0	702	32·7
J. P. Coen,	1915	11,693	15,600	2	6	27-44-75	49	85·0	693	32·4

* Approximate figures.

Cargo Steamers (Reciprocating Engines).

Boilers.				Trial Results.		Weight of Machinery.		Trial I.H.P. per				Weight per Ton of Displacement.
Number and Kind.	Steam Pressure.	Grate Area.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery	G.R.T.	Sq. Ft. of Grate.	100 Sq. Ft. of T.H.S.	
	Lbs.	Sq. Ft.	Sq. Ft.		Knots.	Tons.	Lbs.					
2 D.E. & 2 S.E. cylindrl.,	150	192	7,000	2,320	12·0	528	505	4·40	0·624	12·14	33·1	·0680
,,	180	216	7,392	2,900	13·0	533	412	5·44	0·600	13·43	39·2	·0500
2 D.E. ,,	160	192	6,750	3,036	14·0	508	373	6·00	1·050	15·80	44·9	·0980
,, ,,	180	346	11,512	3,800	13·0	795	481	4·76	0·577	10·98	33·0	·0550
S.E. ,, F.D.	200	382	18,549	8,300	15·8	1,300	352	6·38	0·968	21·73	44·8	·0730
,, ,	180	394	13,298	4,610	13·0	840	408	5·49	0·456	11·70	34·7	·042
D.E. ,,	210	344	11,660	3,350	14·0	754	504	4·44	0·788	9·74	28·6	·115
3 S.E. ,,	200	182	6,414	2,430	11·0	434	400	5·60	0·456	11·46	32·6	·039
,, ,,	210	593	21,100	7,310	15·0	1,528	468	4·78	0·580	12·33	34·7	·061
D.E. ,, F.D.	210	253	9,750	3,810	14·5	769	452	4·95	0·847	15·10	38·5	·098
2 D.E. ,, F.D.	180	220	9,610	5,356	15·3	974	407	5·50	0·950	25·2	55·7	·080
2 D.E. & 2 S.E. . F.D.	200	306	13,080	4,500	14·5	897	446	5·02	0·657	14·70	34·4	·065
S.E. ,, F.D.	210	570	25,803	9,000	16·0	1,985	490	4·58	0·651	15·90	35·3	·088
,, ,,	210	485	17,100	5,200	13·0	1,185	510	4·39	0·400	10·72	30·4	·047
8 D.E. ,,	210	1,014	41,680	13,500	17·0	2,975	493	4·54	0·600	12·33	30·0	·099
2 D.E. & 2 S.E. ,, F.D.	200	314	12,500	4,800	12·0	900	420	5·33	0·608	15·30	38·4	·041
3 D.E. & 3 S.E. ,, F.D.	180	557	22,464	9,800*	17·0	1,790*	410	5·46	1·292	17·60	43·6	·143
7 D.E. ,, F.D.	215	826	31,800	12,500*	16·0	2,735*	490	4·57	0·735	15·13	39·3	·107
2 D.E & 3 S.E. ,,	215	657	25,047	8,100*	15·0	1,845*	510	4·39	0·701	12·33	32·4	·100
8 D.E. & 4 S.E. ,, F.D.	215	1,116	46,436	16,500*	17·0	3,604*	490	4·57	0·683	14·79	35·6	·115
6 D.E. ,,	220	1,019	38,982	15,000*	17·5	3,350*	500	4·48	1·125	14·72	38·5	·167
,, ,,	210	1,086	40,100	12,349	16·6	2,850	512	4·33	0·680	11·39	30·8	·118
S.E. , S.H.	220	240	11,600	4,100	12·7	1,008*	550*	4·07	0·566	17·1	35·3	·082
3 D.E. ,, F.D.	200	450	18,900	6,943	14·5	..	..	..	..	15·4	36·7	..
3 D.E. & 2 S.E. ,, F.D.	200	460	19,412	7,000	15·0	1,470*	470	4·77	0·728	15·2	36·1	·104
2 D.E. & 2 S.E. ,, F.D.	215	440	18,188	5,358	13·5	1,360	566	3·94	0·482	12·2	29·4	·0687
8 S.E. , F.D.	210	520	21,175	6,800	15·0	1,410*	465	4·82	..	13·1	32·4	·0904

TABLE LXXXVIII.—Examples of Small Passenger-

Name of Ship.	When Built.	Tonnage.		No. of Screws	Cylinders.			Revolutions per Min.	Speed of Piston.	Referred Mean Pressure.
		Gross Register	Displacement.		No.	Diameters.	Stroke.			
		Tons.	Tons.			Inches.	Ins.		Feet.	Lbs.
Lincoln, . . .	1882	1,075	2,300	1	2	36-69	39	71·0	462	26·5
Retford, . . .	1882	961	1,600	2	4	23-44	27	108·0	486	23·4
Cambridge, . . .	1885	1,259	1,970	2	4	30-57	36	89·0	534	23·8
Eldorado, . . .	1886	1,514	2,200	1	3	28-43-70	39	88·0	572	31·2
Gazelle, . . .	1889	672	790	2	6	16·5-26-41	30	121·0	605	34·1
Blarney . . .	1889	1,250	2,270	1	3	24-38-62	42	83·0	581	32·5
Lydia, . . .	1890	1,059	1,870	2	6	24-37-56	33	165·0	907	44·0
Duke of Clarence, . .	1892	1,458	1,780	2	6	22-34-51	33	150·0	825	39·8
Bruno, . . .	1892	841	1,775	1	3	23·5-35-57	33	104·0	572	34·6
Chelmsford, . . .	1893	1,635	2,250	2	6	26-39·5-61	36	130·6	780	31·8
Ibex, . . .	1895	1,062	1,270	2	6	22-34-51	33	152·0	836	41·0
Vienna, . . .	1995	1,753	2,342	2	6	26-39·5-61	36	134·0	804	38·2
Roebuck, . . .	1897	1,186	1,740	2	6	23-36-56	33	160	880	47·2
Prince Edward, . .	1897	1,414	1,260	2	6	19-30-48	24	192	768	36·7
Munster, . . .	1897	2,632	2,850	2	8	29-45-48-48	33	175	96	45·2
Duke of Cornwall .	1898	1,540	2,368	2	6	22·5-34-38·5-38·5	33	160	880	44·4
Duke of Devonshire, .	1898	1,205	1,720	2	6	21-32-50	30	175	875	48·1
Dresden, . . .	1898	1,839	2,400	2	6	26-39·5-63	36	135	810	36·9
Prince George, . .	1899	2,041	2,100	2	8	26-40-45-45	30	177	885	36·2
Princess Mary, . .	1900	..	2,147	2	6	23·2-37·4-59·5	36	122·5	735	34·5
Anglia, . . .	1901	1,863	2,300	2	8	26-40-43-43	33	175	962	42·4
Colchester, . . .	1901	1,160	1,970	2	8	14·5-23-27-27	33	145	798	44·6
Antrim, . . .	1905	2,100	2,150	2	8	23-36-42-42	30	195*	955	44·0
Princess Juliana, . .	1909	2,885	2,760*	2	8	28-43·5-49-49	33	170*	935	40·2
Patriotic, . . .	1912	2,254	3,500*	2	8	21·5-35-41-41	36	160*	960	42·0
Bernstorff, . . .	1913	2,310	3,140	1	4	27-44·5-52-52	42	110	770	38·4
Jupiter, . . .	1916	..	3,875	1	3	26-42-69	42	106	742	36·9

* Approximate figures.

Cargo Steamers (Reciprocating Engines).

Boilers.				Trial Results.		Weight of Machinery.		Trial I.H.P. per				Weight per Ton of Displacement.
Number and Kind.	Steam Pressure.	Grate Area.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery	G.R.T.	Sq. Ft. of Grate.	100 Sq. Ft. of T.H.S.	
	Lbs.	Sq. Ft.	Sq Ft.		Knots.	Tons.	Lbs.					
2 S.E. cylindrical,	85	100	3,450	1,390	12·7	233	376	6·00	1·29	13·9	40·3	·101
,, ,,	85	83	2,700	1,065	13·0	174	366	6·12	1·12	12·83	39·4	·109
2 D.E. ,,	85	220	5,820	2,373	15·03	386	364	6·15	1·88	10·79	40·8	·196
2 D.E. ,, F.D.	154	140	5,620	2,082	15·00	293	315	7·10	1·45	14·87	37·1	·133
2 S.E. ,, F.D.	150	118	3,800	1,650	17·00	250	339	6·60	2·45	14·00	43·4	·372
2 D.E. ,,	160	126	4,080	1,550	13·00	289	418	5·36	1·24	12·30	38·0	·123
2 D.E. ,, F.D.	160	320	11,956	5,977	19·50	530	200	11·09	5·64	18·7	49·0	·283
2 D.E. , F.D.	160	258	8,000	4,000	18·50	420	235	9·51	2·75	15·5	50·0	·236
2 S.E. ,,	150	114	3,920	1 530	14·25	192	281	7·97	1·82	10·20	39·2	·108
5 S.E. ,,	160	357	11,450	4,400	18·00	593	302	7·42	2·69	12·32	38·4	·204
2 D.E. ,, F.D.	160	250	8,000	4,250	19·00	420	221	10·12	4·00	17·00	53·1	·331
5 S.E. ,, F.D.	160	357	11,200	5,450	18·00	587	241	9·29	3·11	15·27	48·6	·251
2 D.E. ,, F.D.	175	250	10,850	6,200	20·00	603	218	10·27	5·23	24·8	57·4	·347
2 D.E. ,, F.D.	190	170	6 200	3,100	18·80	304	220	10·20	2·19	18·23	50·0	·217
4 D.E. ,, F.D.	160	520	18,500	9,500	23·8	850	200	11·17	3·61	18·5	51·3	·309
4 S.E. ,, F.D.	180	317	10,772	5,520	19·8	612	248	9·02	3·59	17·40	51·1	·249
2 D.E. ,, F.D.	160	257	8,900	5,000	19·5	505	226	9·90	2·63	19·40	56·2	·294
5 S.E. ,, F.D.	165	357	11,160	5 641	18·5	590	234	9·56	3·06	15·8	50·5	·246
4 S.E ,, F.D.	180	280	11,440	5,892	20·3	568	212	10·56	2·89	21·70	52·6	·227
2 D.E. ,,	155	318	11,320	4,250	17·9	439	231	9·70	1·34	13·40	37·5	·205
8 S.E. ,, F.D.	160	430	15,000	7,170	21·0	675	210	10·7	3·85	16·67	47·8	·294
3 S.E. ,,	180	218	6,520	2.377	16·1	385	363	6·2	2·05	10·90	36·5	·184
2 D.E & 1 S.E. ,, F.D.	150	402	12,460	7,200	21·9	826	257	8·72	3·43	17·9	57·8	·384
4 D.E ,, F.D.	190	534	19,450	9,350*	22·5	890*	200*	10·50	3·24	17·5	48·1	·322
2 D.E. & 1 S.E. ,, F.D.	195	386	14,437	6,400*	18·0	600*	210*	10·67	3·10	16·6	44·3	·171
	210	..	..	3,800	16·0	..	..	..	1·64	..	..	..
3 S.E. ,, F.D.	190	180	7,500	3,100	14·5	..	..	..	..	17·2	41·3	..

TABLE LXXXIX.—Particulars of Paddle-Wheel Steamers' Machinery.

Name.	When Built.	Tonnage.		Kind of Engine.	Cylinders.			Number of Revolutions.	Speed of Piston.	Referred Mean Pressure.	Boilers.				Trial Trip.		Weight of Machinery		Trial I.H.P. per				Weight of Machinery. ÷ Displacement.
		Gross Register.	Displacement.		Number.	Diameter.	Stroke.				Number and Kind.	Pressure.	Grate.	Total Heating Surface.	Total I.H.P.	Speed.	Total.	Per I.H.P.	Ton of Machinery.	Ton Gross Register.	Square Foot of Grate.	100 Feet of Total Heating Surface.	
						Ins.	Ins.		Ft.	Lbs.		Lbs.	Sq. ft.	Sq. ft.		Kts.	Tons.	Lbs.					
Mirimaree,	1871	..	..	Oscillating,	2	80	84	29·0	406	23·8	Box,	30	450	12,500	3,248	18·0	495	341	6·56	..	7·22	25·9	..
Osborne,	1872	..	..	Do.,	2	80½	84	28·6	400	25·0	Do.,	30	412	11,000	3,373	16·6	516	342	6·54	..	8·18	30·7	..
Mary Powell,	1872	..	800	Beam,	1	72	144	23·0	552	22·0	....	30	152	2,660	1,490	17·2	..	..	..	..	9·80	56·0	..
Vigilant,	1872	..	..	Oscillating,	2	64	54	35·0	315	30·8	Box,	30	225	6,032	1,750	14·5	245	314	7·14	..	7·78	29·1	..
Bosphorus,	1874	..	299	Compound Oscil.,	2	34-57	42	51·0	357	23·5	2 S.E. cylindrical,	60	78	1,928	650	14·0	121	417	5·37	..	8·33	33·6	·410
Scutari,	1875	..	..	Do. Diag.,	2	28-52	42	40·0	280	34·7	1 do. do.,	100	62	1,500	625	13·1	96	345	6·50	..	10·10	41·7	..
Bramapootra,	..	..	..	Do. do.,	2	22-37½	30	56·0	280	30·9	2 Locomotive,	90	40	1,058	288	12·5	38·3	298	7·57	..	7·20	27·2	..
South America,	..	..	..	Do. Oscil.,	2	46-82	60	31·6	316	27·9	4 S.E. oval,	70	160	4,310	1,410	12·7	269	427	5·24	..	8·81	32·7	..
Russia,	..	..	..	Do. Diag.,	2	35-62½	51	34·0	289	27·6	2 do. do.,	70	86	2,328	753	12·8	137	407	5·50	..	8·75	32·6	..
P. Marie,	1877	1606	..	Do. Oscil.,	2	60-104	84	32·7	458	30·0	4 do. cylindrical,	70	364	7,304	8,543	17·1	..	..	..	2·21	9·74	48·5	..
Invicta,	1882	1236	..	Oscillating,	2	80	78	33·8	439	29·5	4 Box,	34	604	14,390	3,967	17·7	522	295	7·60	3·21	6·57	27·5	..
Ireland,	1885	1952	..	Do.,	2	102	102	27·1	461	30·6	8 do.,	30	807	22,750	7,000	23·0	1050	336	6·67	3·60	8·75	30·8	..
Empress,	1887	1213	..	Compound Diag.,	2	58-104	72	45·3	544	43·1	4 D.E. cylindrical,	110	407 F	13,720	6,000	19·5	..	..	..	4·94	14·74	43·8	..
Grimsby,	1889	351	402	Do. do.,	2	35-60	48	47·0	376	30·5	3 Gunboat,	80	102 F	2,700	1,105	15·0	148	300	7·47	3·15	10·83	40·9	·369
Violet,	1890	1175	1485	Triple Stpl.,	3	44-70-108	78	35·0	455	32·2	6 Locomotive,	150	282 F	10,022	4,070	19·5	550	302	7·42	3·47	14·43	40·7	·370
Neptune,	1892	378	..	Compound Diag.,	2	33-62	60	57·0	570	38·5	2 Gunboat,	115	..	..	2,000	18·5	..	..	..	5·00	..	..	..
Marie Henriette,	1893	1451	1850	Do. do.,	..	60-108	84	52·9	740	39·6	8 S.E. cylindrical,	117	465 F	14,100	8,130	22·2	..	..	..	..	..	..	..
La Marguerite,	1894	2205	1868	Do. do.,	2	56-110	72	52·0	624	44·2	4 D.E. do.,	140	616 F	..	7,500	22·3	..	..	..	3·40	12·18	..	..
Priscilla,	1894	5292	5200	Do. do.,	4	51-95	132	24·5	539	40·4	10 S.E. do.,	150	850	34,700	9,345	19·0	..	..	..	1·76	11·00	27·2	..
Duchess of York,	1896	996	..	Do. do.,	3	08-48-68	72	43·0	516	37·8	4 do. do.,	120	270 F	8,416	4,300	17·0	455	237	9·45	4·31	15·91	51·2	..
Paris,	1896	2000	..	Triple do.,	3	30-49-75	84	32·0	448	36·0	4 do. do.,	..	140 F	5,150	2,200	16·0	..	..	..	1·10	15·70	42·7	..
Essex,	1896	297	330	Compound do.,	2	24-48	42	55·0	385	36·9	2 do. do.,	110	81 F	2,260	780	14·0	104	298	7·50	2·61	9·03	34·2	·315
Alicia,	1896	230	260	Do. do.,	2	20-40	48	50·0	400	31·3	2 Locomotive,	130	..	2,400	600	12·0	76	283	7·90	2·61	..	25·0	·304
Le Nord,	1898	1700	1850	Triple do.,	3	42-59-87	88	52·0	763	51·0	Water-tube,	186	516 F	16,146	7,000	21·5	..	..	..	5·34	13·56	43·2	..
Empress Queen,	1897	2140	2500	Compound do.,	3	92-68-92	84	50·0	700	40·6	4 D.E. cylindrical,	140	652	26,000	11,440	21·7	..	..	..	5·32	17·50	44·0	..
Detroit III.,	1915	6061	6370	Do. do.,	3	92-62-92	102	28·3	481	39·3	6 S.E. & D.E. oval,	160	506	23,700	7,610	18·3	..	..	..	1·25	15·00	32·1	..

TABLE XC.—EXAMPLES OF TURBINE-DRIVEN SCREW STEAMERS.

Description of Ship.	Principal Dimensions. Length, Beam, Draught of Water.	Tonnage.		Arrangement of Turbines.	Trial Results.			Weight of Machinery per S.H.P.	Propellers.			S.H.P. per	
		Gross Register.	Displacement.		Speed.	S.H.P.	Revs. per Min.		No. of	Diameter.	Pitch.	Ton of Machinery.	100 Ft. of T.H.S.
	Feet.	Tons.	Tons.		Knots.			Lbs.		Ft.	Ft.		
Battleship, . . .	575 × 95 × 28·5	..	27,500	Geared.	20·53	23,500	213	..	2	14·50	12·0	..	53·1
,, . . .	554 × 93 × 28·5	..	26,000	Direct.	21·45	32,126	321	..	4	10·00	8·2	..	..
,, . . .	510 × 88 × 27·9	..	21,200	,,	22·08	40,510	364	..	4	9·20	8·5	..	..
,, . . .	510 × 86 × 27	..	20,000	,,	21·50	26,600	320	179	4	..	..	12·6	..
Battle cruiser, . .	656 × 93·5 × 27	..	24,600	,,	28·13	89,740	..	..	4	..	..	..	..
Cruiser, . . .	430 × 47 × 15·3	..	4,800	,,	26·30	24,335	500	96	4	..	..	23·2	..
,, . . .	360 × 40 × 14·5	..	3,000	,,	23·60	14,200	450	84	3	6·68	6·59	26·5	54·6
T. B. Destroyer, . .	310 × 29·7 × 9·4	..	1,090	,,	29·50	18,625	560	51	2	7·83	7·00	43·9	82·4
,, . .	289 × 26 × 8·30	..	743	,,	30·40	12,800	630	52	2	6·60	6·30	42·7	71·1
,, . .	250 × 25·6 × 8·75	..	735	,,	32·25	15,700	675	51·5	2	7·10	6·50	43·5	75·0
,, . .	267 × 28 × 8·2	..	930	,,	27·10	14,500	750	56	3	5·50	5·00	40·0	..
Atlantic Express, . .	870 × 97 × 35	45,650	51,500	,,	23·45	56,000	190	360	4	16·50	14·50	6·22	40·4
,, . .	760 × 87·5 × 30	32,500	36,440	,,	26·40	72,400	192	308	4	17·2	16·13	7·29	45·7
,, . .	650 × 72·4 × 30	19,500	27,500	,,	20·2	24,000	185	366	3	14·0	12·75	6·13	46·1
,, . .	520 × 60 × 28·5	10,629	17,850	,,	19·50	12,000	325	312	4	8·75	7·20	7·18	46·6
,, . .	548 × 66 × 30·5	14,350	22,000	Geared.	17·65	10,900	137	349	2	15·25	15·00	6·41	35·5
Ocean Express, . .	500 × 63 × 21·0	8,255	9,700	Direct.	23·99	25,500	357	300	4	..	8·00	7·47	42·5
Channel Express, . .	375 × 46 × 13·4	2,651	3,353	,,	23·27	14,700	466	200	3	7·17	6·67	11·20	53·6
,, . .	350 × 42 × 11·0	1,951	2,400	,,	23·53	9,500	430	..	3	6·70	6·60	..	48·2
,, . .	330 × 43 × 11·7	2,174	2,270	,,	23·14	8,500	520	168	3	6·70	5·60	13·33	63·4
,, . .	310 × 40 × 9·9	1,680	1,750	,,	23·94	9,000	440	..	3	6·70	6·50	4·54	75·0
,, . .	295 × 34·5 × 9·5	1,646	1,485	,,	24·06	10,000	670	156*	3	5·42	4·74	14·36	46·8
,, . .	284 × 39 × 12·0	1,498	1,990	,,	20·00	6,670	500	200*	3	5·50	5·10	11·20	51·3
,, . .	290 × 36 × 12·0	1,567	1,864	Geared.	19·70	4,980	300	140*	2	8·00	8·00	16·0	48·70
,, . .	275 × 38·5 × 9·75	2,163	1,800	Hydraulic.	20·06	5,330	453	..	2	6·56	5·58	..	43·60
,, . .	295 × 35·5 × 9·0	1,774	1,510	Geared.	25·07	12,300	435	160*	2	..	7·00	14·0	49·92

* *N.B.*—Water-tube boilers.

CHAPTER XXVIII.

EFFECT OF WEIGHT—INERTIA AND MOMENTUM—BALANCING THE SAME.

HITHERTO in this book little or no consideration has been given to the effect on the engine of the weight of any of its parts; and although that of each one has some effect on itself and on its immediate surroundings, it may generally be very trifling. The effect of the weight of some of the parts, however, is perceptible throughout the whole engine, and sometimes even makes its presence known from stem to stern of the ship. In the case of the propellers, both screw and paddle, their weight has been taken into account in estimating the stresses on their shafts and the pressure on the bearings supporting them; also the weight of the shafts themselves has not been overlooked in considering the dimensions of their journals, bearings, etc. But in calculating the sizes of shafts and propellers no reference has been made to the fact that their weight does to some extent modify the stresses borne by their parts. As a rule, however, the effect of mere weight is so very small compared with that due to the steam pressure on the pistons, cylinder covers, etc., that in ordinary engines it is and may be generally neglected. But the great increase in rate of revolutions in modern engines, and the demand for greater perfection in working, as well as the absolute necessity for it in many cases, have drawn more particular attention to the subject; some most interesting investigations have been made by scientists, and from time to time methods devised whereby the ill effects due to the inertia of the moving parts of an engine may be modified or removed altogether, so that it may be run at any speed without distress of any kind to itself or surroundings, and with perfect freedom from vibration. In modern design it is now unusual to neglect the effects of inertia.

Fixed Parts—that is, those parts the centre of gravity of which have no motion but that due to the motion of the ship—may first be taken into consideration. The effect of their weight is simply that due to gravity, and acts always vertically. For example, the cylinder and its appurtenances of a vertical engine rest on the columns, and by their weight increase the load on the columns; on the other hand, the stress on the bolts connecting them to the columns is decreased by it. When, however, the ship is in motion they have momentum, so that in case of a sudden stoppage by grounding or collision the tendency would be for them to move forward and shear the bolts, etc., and to bend the columns as well as strain the column bases and connections to the ship. The tendency is also of the same nature when the ship rolls heavily, but in this case motion is transverse to the forward movement. It is usual and advisable in all vertical engines, especially in tall ones, to provide for these contingencies by fitting ties of a more or less flexible nature from the cylinders to a suitable part of the ship. In a minor degree the same remarks are applicable to the columns, especially if of cast iron and heavy, but being lower down and sharing in the external support given to the cylinders by the ties, there is no need to do more than brace them together, as is generally done when the columns are of wrought iron or steel.

By the above definition the shafting is a "fixed part," whose weight sets up in itself bending moments as well as shearing forces which may be sufficiently large to take into account with the twisting moment if the bearings are far apart. A rotating shaft has also critical speeds of whipping, at which the centrifugal forces become unbalanced; and although in the case of marine shafts dangerous stresses may not necessarily be set up, the result is a considerable amount of vibration and racking on the bearings. As a rule, greater stability is secured at speeds above the critical speed, but in marine practice it is perhaps best, if possible, to arrange for the critical speed to be above the maximum engine speed.*

Moving Parts.—The effect of weight in these, however, demands the chief attention, and in quick-running engines the greatest attention of the designer, not so much, as will be seen, for its static as for its kinetic effect. Some of these have simple rectilineal motion, others have circular motion, while a few have a combination of both, and move in an elliptic orbit; others, again, reciprocate on curved lines. The static effect of the weight of the piston, piston-rod, crosshead, and connecting-rod, is to increase the load on the piston on the downstroke and decrease it on the upstroke, in both cases uniformly. The effect of the weight of the crank-arms and pin (if unbalanced) is also to increase the down and decrease the up load, but not uniformly in this case. The effect of the former is generally measured by dividing the weight in pounds by the area of the piston in square inches, and adding the result to the effective mean pressures of the steam on the top of the piston and deducting it from that on the bottom. This is shown graphically by drawing the diagram of effective pressures of the top above a reference line and that of the bottom below it: a new reference line is now drawn below the original at a distance representing the pounds thus ascertained and the ordinates measured from it used to determine the load on rods, etc., and the twisting moments on the crank-shaft.

Fig. 280 is the indicator diagram from the high-pressure cylinder of a triple-compound engine 31 inches diameter; the weight of the piston, piston-rods, connecting-rods, etc., is 5,728 lbs.; dividing this by 755 the

* The critical speed (R.P.M.) of whipping of a shaft of diameter d (feet) and length L (feet) is

$$N = 393{,}000 \frac{d}{L^2} \text{ for shaft } \textit{supported} \text{ on bearings.}$$

Example.—A shaft 10 inches (0·834 ft.) diameter is supported on bearings 25 feet apart—

$$N = 393{,}000 \times \frac{\cdot 834}{625} = 524 \text{ R.P.M.}$$

If the shaft be regarded as *fixed* in the bearings,

$$N = 2{\cdot}28 \text{ times the above value.}$$

For more than two equidistant bearings, the above formulæ still hold, but for bearings unequally spaced the problem is complicated. These formulæ will generally indicate what speeds to avoid. The advanced mathematical reader may consult Dr Chree on "Whirling of Shafts," *Proc. Physical Soc.*, vol. xix.

area of the piston in square inches, the constant downward pressure due to the weight is equal to 7·6 lbs. per square inch. The figure in plain lines is that taken from the top, that in the dotted lines from the bottom of the

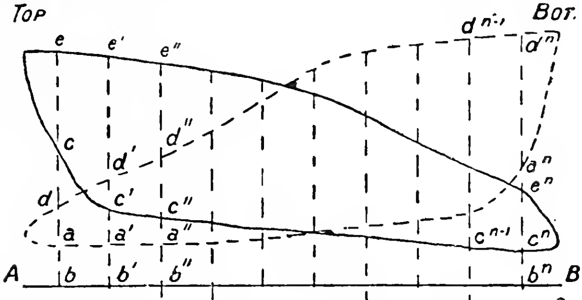


Fig. 280.—High-pressure Cylinder Diagram.

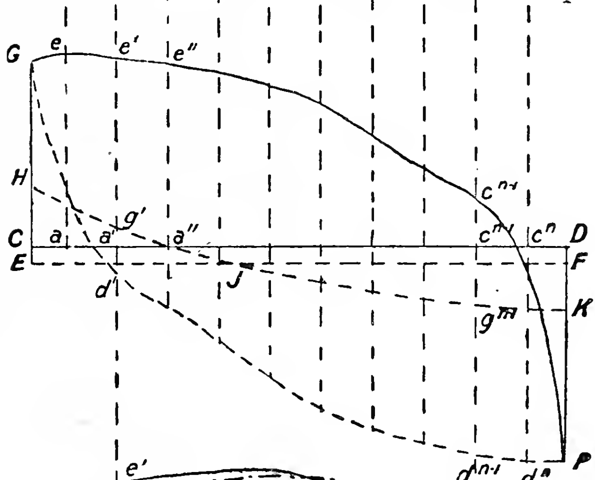


Fig. 281.—Diagram of Effective Pressures Neglecting Inertia.

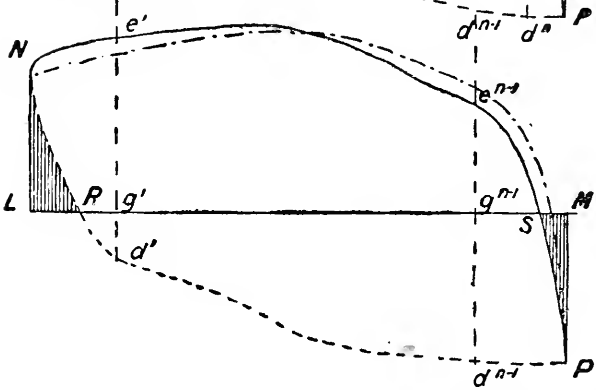


Fig. 282.—Diagram of Effective Pressures including those due to Inertia.

cylinder. A B is taken as a base line divided into any number of parts at points b b', etc., through which ordinates b e, b' e', etc., are drawn, and of which a e, a' e', etc., represent the effective pressure on the top of the piston when in those positions on the downstrokes, and *c d*, *c′ d′*, etc., the effective pressures on the underside of the upstroke.

Now take C D as a base line (fig. 281) of the same length as A B; divide in the same way and draw a diagram by taking ordinates through the points, cutting off *a e*, *a′ e′*, etc., the same length as *a e*, *a′ e′*, etc., in fig. 280, etc. On the lower side construct in a similar way the figure P d^n *d′* G from the dotted diagram, fig. 281.

Now take a point E below C, so that C E represents on the same scale as the diagrams the pressure per square inch on the piston equivalent to the weight of the piston, etc., and draw E F parallel to C D. Then any ordinates on E F, intercepted by G *e e′* P, etc., represent the true effective pressure on the downstroke, and similarly any ordinates from E F downward, intercepted by P *d′* G represent the true effective pressure on the upstroke at those points. The maximum ordinate on the lower side indicates the tension on the connecting-rod and piston-rod bolts for purposes of calculation, and the maximum ordinate on the top side of E F indicates the compressive load on the connecting-rod, etc., for the same purpose; the actual load being obtained. of course, by multiplying the number of pounds which these ordinates represent by the number of square inches of piston area. The curve of twisting moments of a vertical engine should also be obtained by using these ordinates.

A further modification may be made in the case of engines having the air and circulating pumps worked by means of levers, as then the weight of the moving parts of the pumps and their loads tend to balance the weight of the piston. Balance weights on the cranks opposite the arms have also been fitted with the object of balancing the *weights* of pistons, etc., and have also very properly been fitted with the sole object of balancing the crank weight only.

The weight of the slide valves and valve motions also modify in a similar way the stresses on their parts but in a minor degree.

In horizontal engines the pistons, rods, etc., do not by gravity affect the load on the pistons, but the weight of the crank-arms and part of the weight of the connecting-rod do affect the turning moment, but in a different way from that of a vertical engine. In their case the forward effort of the piston was resisted by the weight of crank, etc., till a little past the half-stroke and assisted from the piston to the end of the stroke; on the back-stroke the assistance continued till nearly half-stroke, when it ceased, and resistance set up and increased to the end. Here balance weights could be, and were, used to advantage, so that most of the horizontal engines had them, while a vertical engine was rarely seen with these appendages.

Momentum.—After all is said, the effect of mere weight or gravity on the parts of an engine is very limited and never serious—in fact, may generally be neglected as has been shown. But the forces set up by heavy moving parts when in motion—especially in rapid motion—are quite different both in magnitude and effect. To overcome the inertia of a heavy body and start it into motion a considerable force is required; to accelerate the motion still more force is required. A heavy body when in rapid motion has much energy stored in it, all of which must be abstracted in order to bring it to rest; if the process of bringing to rest is accomplished quickly the force developed will be large and produce disastrous results, unless special provision is made for otherwise absorbing it. This force varies with the mass of the body and its acceleration; hence in a quick-running engine the inertia forces set up in the moving parts are much more serious matters than in a slow-moving engine, not only on account of the stresses set up in the parts themselves and in those adjacent to them, but because of the vibratory effects on the structure of the engine generally, as also on the structure by which it is carried. Inertia forces can be modified by reducing the weight of a moving part, but the velocity is practically fixed, being that required for the full working of the engine. Very little scope is permitted. therefore, to the designer to modify the motion of any one part.

Dealing first with those parts which have simple linear motion, such as the piston, piston-rod, and crosshead, the valves and valve-rods, etc., the general effect can be appreciated by tracing each phase of movement. For example, at the commencement of the stroke when steam is admitted to the cylinder, it exerts a pressure on the piston and cover alike, hence the thrust on the piston (and from it to the bearings) and the resistance of the engine framing are equal but opposite in direction, and, consequently, exactly balance one another since there is no motion. There is, therefore, no load on the parts external to the engine. To produce motion, the inertia of the piston, etc., has to be overcome; if this is done by the pressure of the steam, the whole force of the steam on the piston is not transmitted through the rods to the crank, for a considerable portion of it is used up in moving the piston, etc.,* and is thus stored in the piston and its parts as energy; the load on the cover being, therefore, greater than the part transmitted, it tends to make the engine move in the reverse direction to that of the pistons. Further, the movement of the pistons is accelerated in its course by accessions of force till about the half-stroke; hence during all this time the engine is tending to move; in fact, if it is a vertical engine and the piston is on the downstroke, it will jump from its bed unless its weight is greater than the accelerating force, and if there are no holding bolts restraining it. During the latter part of the stroke the momentum of the piston, etc., tends to keep it in motion, and consequently a downward thrust on the rods is created in excess of the load on it and the cover, so that the engine now would move the other way, unless restricted by the bed on which it rests. The same phenomena occur in horizontal engines, and are of the utmost consequence in the locomotive engine; it was, in fact, with this class of engine that the effects of inertia forces were first impressed on engineers by the disastrous results with some outside cylinder horizontal engines when attempts were made to drive them at higher speeds. Fortunately the practical minds of those days soon found a remedy—one that, from its simplicity and efficiency, is still used in locomotive engines, in spite of what science has done to reveal its faults.

The general effect of the variation of momentum of the piston, etc., is to reduce the load on the crank pin at the early part of the stroke, and to increase it at the latter, with the reactive tendency to move the whole engine up and down alternately; in other words, to reduce the ratio of maximum to mean

* In fast-running engines, such as are used for generating electricity (in which the revolutions are constant while the load varies), the inertia of the pistons and rods is so great that on light loads the mere steam pressure on them would not overcome it; in fact, the piston of one engine is started and accelerated by the power transmitted from the others or from the flywheel, even when the load is considerable, hence it is most important that such engines should have good "stern-going" piston-rod guides. If an engine is moved by external means, such as a belt from another engine, the effect of inertia of the moving parts is exactly the same as if it were working under steam.

load, and cause vibration on the bed supporting the engine; as also to reduce the maximum stresses on the rods and their bolts. The vibration set up in the engine bed is, of course, communicated to the ship.

The general effect on the crank of the variation of momentum of the piston-rods, etc., will be dealt with later on, but in the meanwhile can be shown graphically by calculating the accelerating forces applied to the piston at a series of points at and from the commencement of the stroke, and taking as ordinates the equivalent pressures per square inch on the piston; H J K (fig. 281) is such a curve obtained by making E H represent the pressure on the piston required to overcome its inertia, and so also with the other ordinates. If, now, a new base line, L M, fig. 282, is taken and a curve, N e' S P, drawn by taking as ordinates the distances intercepted by the curves H J K and G e e'' P, a true representation of the variation in real push and pull of the rods is shown. The ordinates measure the combined effect of steam pressure on the piston, gravity, and inertia of moving parts. The cushioning is shown by the shaded portions at each end, S M P and R N L. In this case it may be mentioned that the diagram, fig. 280, is that of a three-crank triple-compound engine with the high-pressure crank leading, and to show the difference in effect when the medium-pressure crank leads the high pressure, the dot and dash line (fig. 282) is the curve of equivalent loads for a similar engine under that condition.

The general result of the movement of the slide valves, link motions, etc., is similar, and although their weight is less than that of the piston, etc., and their motion much more restricted, their varying momentum can set up a manifest vibration in a quick-running engine.

The connecting-rod has a complex motion, inasmuch as one end moves in a straight line while the other moves in a circle; its centre of gravity consequently moves in an ellipse whose major axis is equal to the stroke of the piston. Its motion, therefore, may be viewed as partly linear with the piston-rod, etc., and partly circular with the crank. But when in motion it oscillates like a pendulum, so that its effect on the crosshead will be the same as that of a pendulum on its point of support; hence it will set up a horizontal force there in addition to the downward one due to gravity. The force will vary with the mass and the horizontal acceleration; it will set up a horizontal vibration in the engine and increase the stresses on the rod due to cross bending; hence in very fast-running engines the connecting-rod should be made as light as possible consistent with strength. A short rod is, of course, lighter than a longer one, but it gives a greater thrust, due to steam pressure, and, consequently, tends to increase vibration transversely.

The crank, including arms and pin, moves in a circle, and consequently centrifugal forces are set up which tend to press the shaft against the journals in the radial direction of the crank pin for the time being. The weight of the crank-pin brasses and that part of the connecting-rod surrounding them, moves in a circle, and causes further centrifugal forces to act on the crank-shaft. These parts by their momentum also set up tangential forces, which, with the resisting force at the bearings, form a couple tending to keep the shaft in motion; but as the direction changes, so the reacting force changes in direction, and this also tends to rack the bearings and foundation, as well as to set up vibration.

Balancing.—From the above causes a reciprocating engine of any kind when in motion sets up vibrating forces which are only extinguished by doing work of some kind, mostly, however, of a mischievous character. Even the slow-running paddle engine developed the vice, which, when the engine is nearly horizontal and has a single cylinder, as in certain Clyde steamers, is very perceptible fore and aft ways. When the engine was directly coupled to the screw shafting, and its revolutions consequently three or four times that of the older engines, vibration and its effects were soon noticed. Although some general idea of the causes was grasped, and attempts were made to grapple with the evil by fitting balance weights opposite the cranks or in the turning wheel at the end of the crank shaft, no true insight seems to have been gained ; consequently, for more than half a century engineers generally were content to put down the vibration of screw steamers as due to the propeller, and to admit that the paddle wheel was in that respect the superior for passenger steamers.

Now, any force which has to be extinguished by doing useless work is a force wasted, and if the work done is injurious to engine or hull, the loss is worse still. For this reason the engineers of the past are to blame for not having found means to prevent such waste, and still more so, those who now neglect the efficient methods which have been devised for preventing such forces coming into existence, and for rendering them harmless when developed.

Mr. Arthur Rigg called attention to the subject in his able work so far back as 1878, but it is due to Sir A. Yarrow's genius for investigation that the question was put in practical form, and to Mr Otto Schlick's analytical ability that methods were devised whereby balancing was no longer an interesting workshop experiment, but an exact science and a fine art; it has been rendered even more so by the later contributions of Prof. Dalby, Mr. Inglis, and others to the *Transactions of the Inst of Naval Architects, etc.*

To understand clearly what is involved in the balancing of an engine, it will be best to take, first, the common case of a double-acting single-crank engine. Suppose it to have a flywheel with a weight near its rim of such a size and position that at half-stroke it statically balances the crank, rods, and piston. When this engine is in motion the momentum of the principal parts is roughly balanced for turning by that of the balance weight: but a couple is formed by joining the centre of the gravity of the weight by a line going diagonally across the engine to its centre line. The shaft and its bearings will, therefore, be racked by this cross-section couple, and the engine tend to tilt and twist on its bed. Moreover, since the momentum of the balance weight is constant tangentially, while that of the pistons, rods, etc., ceases at the ends of the stroke, there must be an excess momentum of the balance weight at those periods which, in the vertical engine, tends to make it move horizontally, and in the horizontal engine vertically. This evil is made apparent in a locomotive engine by what is called "rail hammering," from the blows the rail gets from the wheels with the balance weights when the engine passes the dead centres.

If the balance weight is placed on one crank-arm only, as was sometimes done for cheapness, similar actions were set up, but less in degree. Two balance weights, one on each arm, make the proper balance, but still fail to avoid the horizontal overbalance when the crank passes the dead points and the consequent horizontal vibration. If only balance weights opposite the cranks can be provided, a compromise must be arrived at, so that their weight is such that the horizontal vibration due to overbalance is reduced to that of the vertical vibration caused by underbalance. If, however, the engine has an air pump, etc., worked by means of levers so that they move in the opposite direction to that of the piston, their weight may be so adjusted that their momentum balances that of the piston and rods, while the balance weights on the crank balance the crank-pin brasses and connecting-rod end. There will, however, still remain a couple tending to rock the engine athwart ship.

Mr. J. H. MacAlpine designed a quadruple engine (fig. 283) with the cylinders in pairs, one behind the other, and each pair operating on the same crank by means of rocking levers like those of the air pump. This gives, no doubt, a very well-balanced arrangement fairly free from vibration, but the ratio of maximum to mean torque is greater than that of a three-crank engine.

Further, the motion of the valve and gear in this single engine will create inertia forces which all tend to set up vertical vibration, and should be taken account of especially if it is a fast-running one. This can be done by finding the resultant forces from all the parts, and placing the balance weights accordingly.

Take now the case of a two-crank engine. If the cranks are opposite one another they will balance for rotation and so require no weights, and if the moving parts are of equal weight their relative momenta will also only tend to make the engine rock in a fore and aft direction on its foundation and not jump on it; it is true, however, that this tendency to rock will be prevented by the holding down bolts, but with the vibration of the ship as a consequence. Such vibration, however, will not be so troublesome as is the case with an unbalanced single engine tending to jump. In this instance the balancing might be effected by a weight in the flywheel at a slight expense, or else by some form of "bob" weight, suggested by Sir A. Yarrow, placed beyond one of the cranks so as by its mass and velocity to balance the momentum on the other crank-pin. This, of course, leads naturally to a three-crank engine with the two outer ones *opposite* the middle crank and the weight of the moving parts of the middle engine equal to those of the two outer engines together, these being of equal weight.

This form of engine would seem to be perfectly balanced, and it would actually be so if it had slot crossheads instead of connecting-rods; as it is, the acceleration of the piston of the middle engine at the commencement of its downstroke is greater than that of the wing engine pistons at commencement of their upstrokes with a consequent loss of balance and vibration.

If a two-crank engine has its cranks at right angles it will be, from the ordinary point of view, a better balanced engine than one with the cranks opposite, although there may be more vibration set up by it. The fact is, that the torque is much more even in this case—that is, the ratio of maximum to mean twisting moment is much less; and the engine runs more uniformly without a flywheel. This little fact enforces the necessity of clearly distinguishing between the balancing for even circular motion and the balancing necessary to prevent vibration either vertical or horizontal, although generally the balancing to prevent vibration tends to improve the even running of the engine. Balancing of these engines may be done by means of a pair of weights, one at each crank, in such a way that the vertical is not extinguished at the expense of excessive horizontal vibration.

The same process must be gone through with a three-crank engine whose

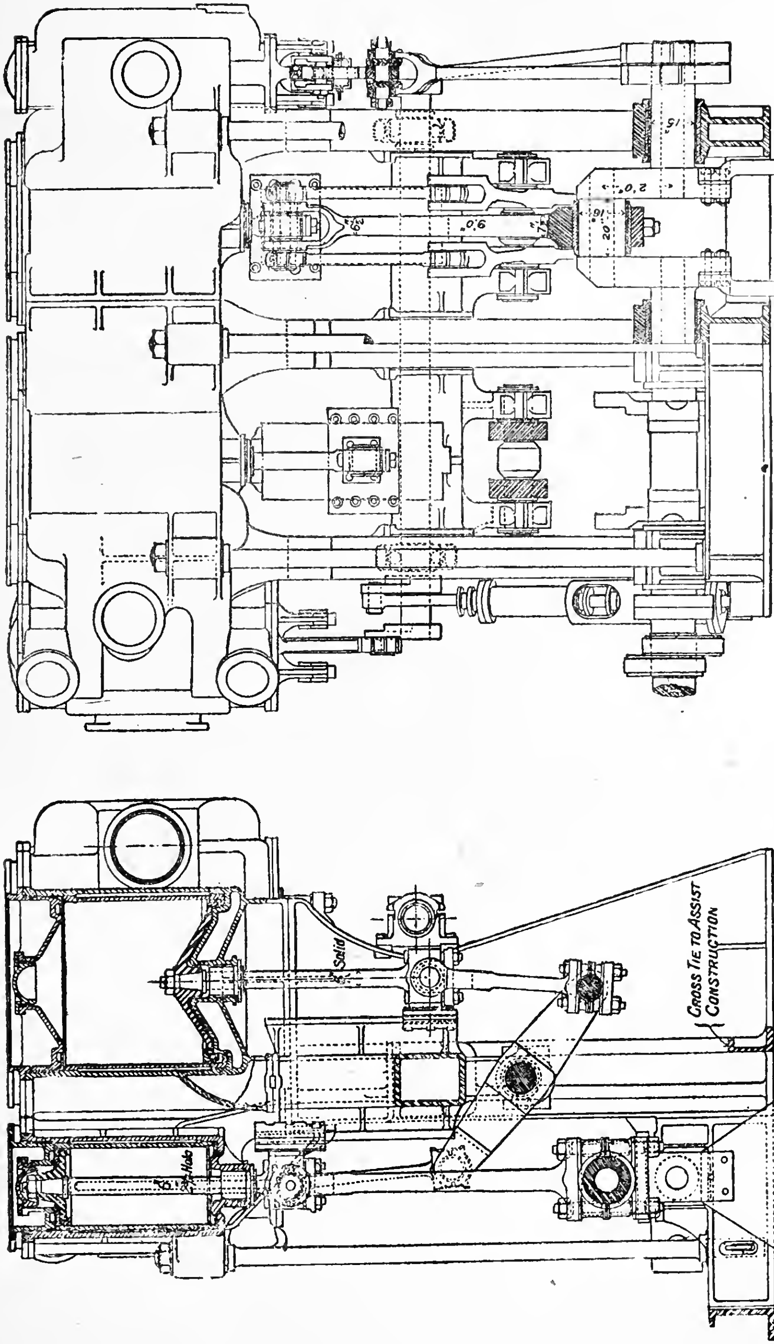


Fig. 283.—MacAlpine's Quadruple Balanced Engine, Cylinders 27″, 39″, 56″, and 80″ diameter × 48″ stroke.

cranks are at an angle of 120° with one another—viz., to ascertain the kinetic resultants of the system, and balance them wholly or partially by means of balance weights placed on the first and third cranks of such a size and position that the engine will neither rock fore and aftways nor jump on its seat; and, further, it must not have excessive horizontal vibration. The methods of arriving at these results are set out fully later on.

A four-crank engine, however, is the easier to deal with, as in that case contiguous engines can have their cranks opposite one another so as to balance vertically and nearly balance horizontally and have the minimum rocking moment. Also, as the extreme cranks are at right angles they give a smaller rocking couple than if the angle were larger. Mr. John Tweedy devised an arrangement of cylinders and cranks in a four-crank triple engine whereby a good balance is obtained without the use of any balance weights or other addition to the engines. In his design the low-pressure cylinders with their gear, as being the lightest, are placed at the ends with the high-pressure and medium-pressure cylinders between them. The first and second cranks are nearly, but not quite, opposite one another, as are also the third and fourth; the second is not quite at right angles with the third; in fact, the cranks are all so set as to give the best balance of the whole. This is as perfect an arrangement as can be obtained without the use of balance weights and is such as to be practically without vibration.

Fig. 72 shows such an engine and fig. 284 the arrangement of the cranks to

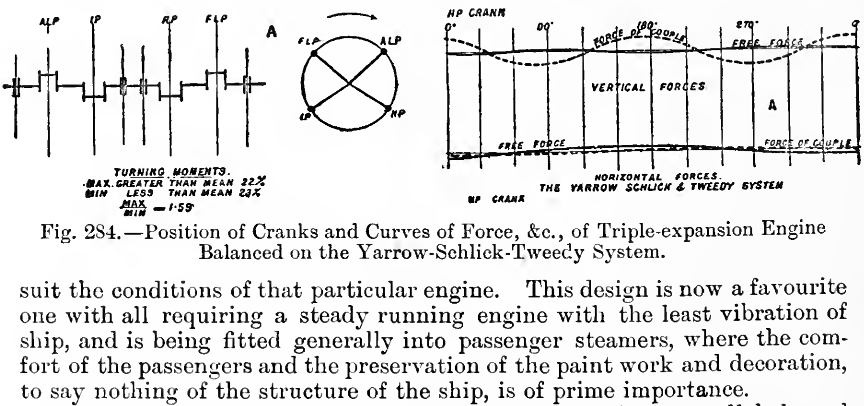


Fig. 284.—Position of Cranks and Curves of Force, &c., of Triple-expansion Engine Balanced on the Yarrow-Schlick-Tweedy System.

suit the conditions of that particular engine. This design is now a favourite one with all requiring a steady running engine with the least vibration of ship, and is being fitted generally into passenger steamers, where the comfort of the passengers and the preservation of the paint work and decoration, to say nothing of the structure of the ship, is of prime importance.

In warships, where accuracy of fire may mean victory, well balanced engines must be fitted. The Admiralty therefore insist on all engines being as perfectly balanced as possible—a point in favour of the turbine.

As, however, a four-crank engine is generally more costly and occupies more space than a three-crank one, and the ratio of its manifold torque (or twisting moment) to the mean torque is greater than that of a three-crank engine, the latter will continue to be largely used. The following investigations and methods of balancing such an engine will be both of interest and importance.

Figs. 285 and 286 show an engine designed and patented by M. E. Wigzell, whereby a nearly perfect balance is obtained by having one cylinder above another with the pistons moving in opposite directions. Here again, the obliquity of the connecting-rods prevents absolutely perfect balance; in practice, however, the engine behaves as if it were perfectly balanced. This engine has the further merit of occupying very little fore and aft space and is readily started from any position.

1. Preliminary Definitions. — The direct object of what is termed "balancing an engine," is to prevent the effect of the motion of its

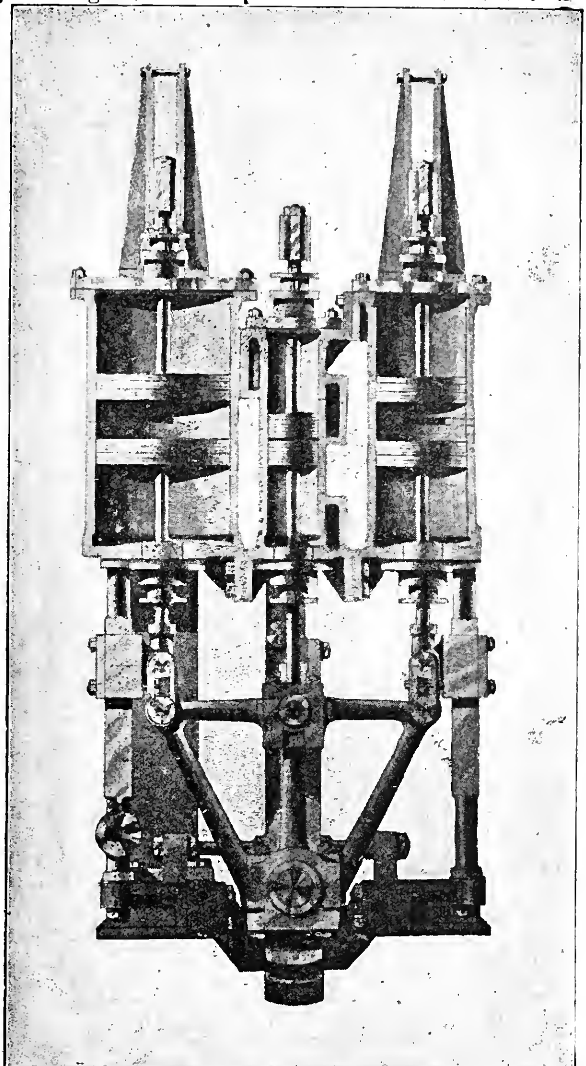


Fig. 285.—Wigzell's Engine. Transverse Section.

moving parts being felt by surrounding objects, and not to eliminate any particular stresses in parts of the engine itself. Neither has the object any connection with fluctuation of speed, the theory and action of flywheels not being related to the subject.

It is a natural *consequence* of balancing an engine that certain stresses are considerably reduced; that others remain as they were, and that some possibly even may be increased.

Another point, which may not at first sight appear evident, is that an engine may be very much out of balance in itself, that is, if placed on a rigid foundation, it causes very severe stresses on the holding-down bolts,

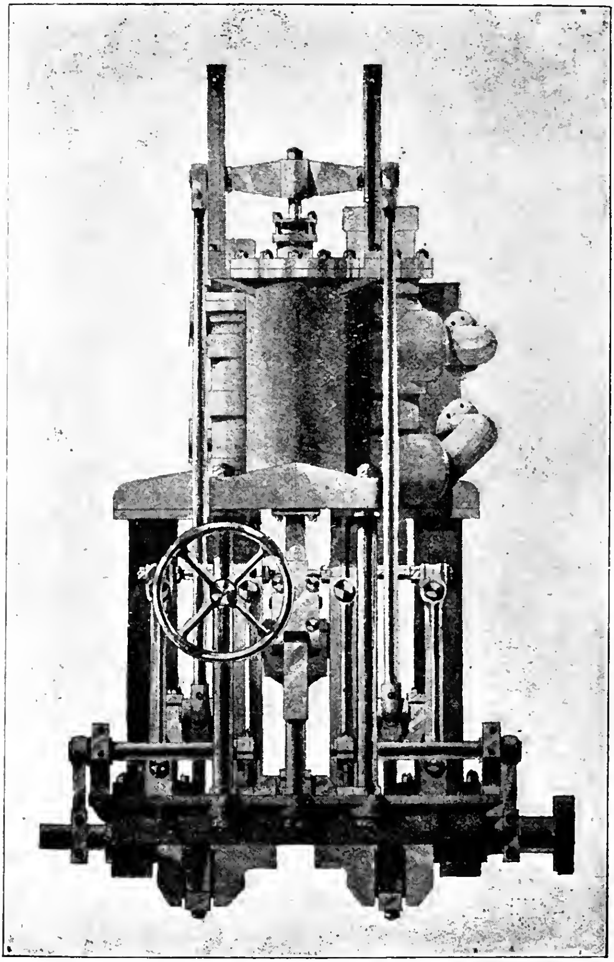


Fig. 286.—Wigzell's Engine. Longitudinal Elevation.

yet if placed in certain positions on a flexible foundation may appear to be in a perfectly balanced condition; that is to say, it will run without producing any vibrations of the foundation, and in fact can run safely without holding-down bolts.

The problem of balancing an engine need not necessarily be dependent on this phenomenon, and very fortunately so, because, although a ship is certainly a flexible foundation for an engine to rest on, so little of the true nature of its flexibility is subject to calculation that, for new ships particularly, any calculations involving this property are really mere guess-work and therefore better left alone.

If, however, the engine be perfectly balanced in itself it can cause no vibration to the foundation, whatever may be the nature of its flexibility.

In practice it is impossible to perfectly balance an ordinary engine and it will be shown here within what limits and to what extent an engine can be balanced. All problems will relate particularly to the vertical engine, but the principles involved apply equally well to horizontal and inclined engines. Before proceeding with this, however, attention must be drawn to the fact that the problem of balancing an engine is not in the least concerned either with steam loads and twisting moments due to the same or with the speed of the engine, the dynamical equilibrium of the moving parts being quite independent of how fast or how slow or how varying the speed of the engine may be. Comparative values of the speeds of the various parts are certainly required, for the magnitude of any resultant forces and couples are determined by the speed, but unless those absolute values are required, only comparative values of the speeds of the parts are essential.

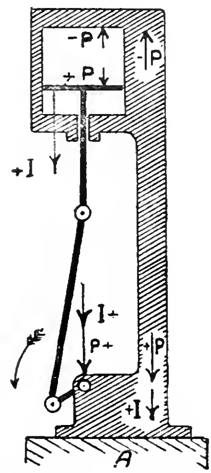


Fig. 287.

As many engineers, some even of high standing, are apt at times to confuse the matter, and imagine that there is some mysterious connection between steam pressures and balance weights, an enlargement of what has been said under "Momentum," Section 2, showing that there is no connection between the two, is given in the following simple demonstration.

Let fig. 287 represent a diagrammatic engine. Let the steam load on the piston be + P. Then the load on the top cover is - P, and, confining attention to the vertical forces, the vertical pressure at the main bearings is + P. That is, the steam load causes no pressure of the engine on the foundation, A.

The moving parts of the engine, however, are not weightless, also they never have a uniform velocity in any given direction (in this case, the vertical)—that is, they always have a certain acceleration, positive or negative. Now, since

$$\text{force} = \text{mass} \times \text{acceleration}$$

another vertical force is consequently introduced, and is—

$$I = \frac{\text{weight of moving parts} \times \text{vertical acceleration}}{g}.$$

Suppose that when the parts are in the position shown, this force equals + I lbs. acting downwards. Since the whole of the moving parts are concerned, this is equivalent to a load + I pressing the piston down as shown. Now it is self-evident that there cannot be any force − I acting on the cylinder cover, therefore this force + I remains alone and its effect is consequently to make the engine heavier by + I lbs. At other points the inertia force will act upwards instead of downwards, and it will also have a continuously varying value.

There are, therefore, up and down forces due to the *weight and velocity of the moving parts only*, acting on the engine as a whole and recurring every revolution.

As these are the forces that tend to make the foundation vibrate, the balancing of an engine consists of counteracting the forces so that the moving parts are really made as if weightless when considered dynamically in certain given directions.

It has also been stated above that the problem of balancing has no direct relation to the speed of the engine. This will be obvious on consideration, for any counteracting weights that may be applied must necessarily have their velocity imparted to them by the engine itself, and the factor of velocity is consequently eliminated from both sides of the representative equations.

2. Harmonic Motion.—The theorem of harmonic motion is stated here because of its great importance and the direct bearing it has on the present subject.

Suppose a point P (fig. 288) to move uniformly round the circle X Y X[1]. Draw two rectangular axes O X, O Y, and P N perpendicular to O Y. Then

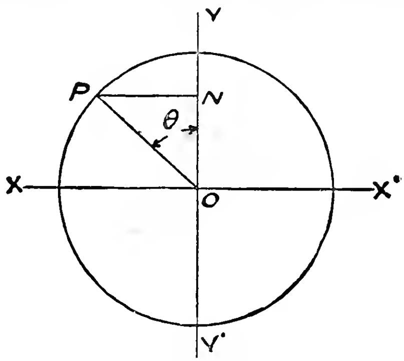


Fig. 288.

if P has a uniform angular velocity N will move along Y O Y[1], with a motion that is called "Simple Harmonic Motion."

Let $\quad O P = r, \qquad P O Y = \theta,$
then $\quad O N = r \cos \theta.$
Let $\quad v =$ linear velocity of P,
then $\quad$ Velocity of $N = -v \sin \theta$
and $\quad$ Acceleration of $N = -\dfrac{v^2}{r} \cos \theta.$

The proof of these will be found in most treatises on theoretical mechanics.

P is called the *generating point.*
The angle θ is called the *phase.*
r is called the *amplitude.*
The time of one revolution of P is called the *period.*

A curve of the displacement of N from O or of the acceleration of N may be drawn if the abscissæ be made proportional to θ and the ordinates pro-

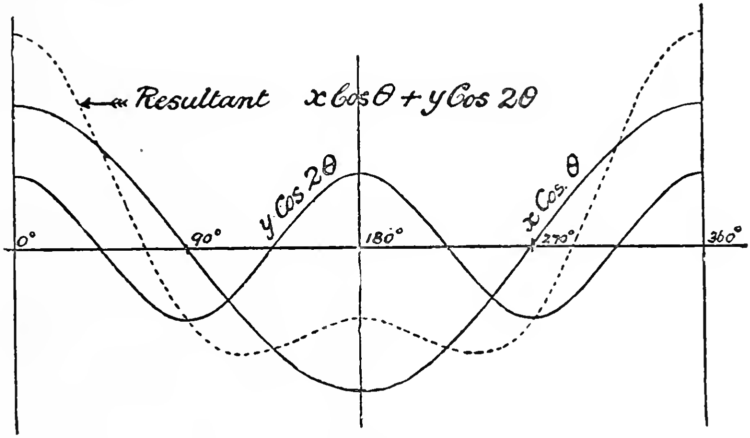


Fig. 289.

portional to the displacement or acceleration. Such a curve is shown in fig. 289) by $x \cos \theta$.

3. To Combine Two or more Harmonic Motions of the same Period—

Let P and Q (fig. 290) be the generating points of two harmonic motions along the axis O Y.

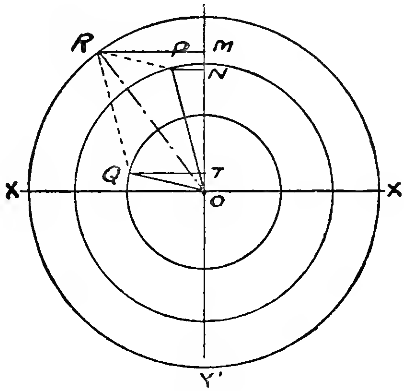


Fig. 290.

Then, since the angular velocities of O P and O Q are the same and are of the same sign, the angle P O Q is constant.

Complete the parallelogram P O Q R. The linear velocities of P and Q are proportional to O P, O Q respectively, and are perpendicular to them.

Therefore O R is proportional to their resultant which is perpendicular to it.

Because O P, O Q and angle P O Q are constant, O R is constant and has the same angular velocity as P and Q. Therefore R is the generating point of the resultant harmonic motion.

The combination of any number of harmonic motions of the same period and sign is analogous to the combination of any number of coplanar forces by the polygon of forces.

4. **The Inertia of the Various Parts of an Engine** will now be dealt with separately.

Inertia of Piston, Rod, and Crosshead.—These fittings all being connected together rigidly, and having only one line of motion, the velocity and acceleration of the whole is the same at all points.

(1) If their movement is controlled by a connecting-rod of infinite length they have a simple harmonic motion, and the acceleration at any angular position θ of the crank from the top centre is therefore

$$f = -\frac{v^2}{r} \cos \theta \text{ (par. 2).} \qquad (1)$$

(2) If their movement is controlled by a connecting-rod of definite length the simple harmonic motion is at once destroyed and the expression for the acceleration is a very complicated one, so much so that it is impossible in practice to use it in its complete form for either algebraical or graphical calculations.

The expression correctly stated in definite form is—

$$f = -\frac{v^2}{r}\left(\cos \theta + \frac{rl^2 \cos 2\theta + r^3 \sin^4 \theta}{(l^2 - r^2 \sin^2 \theta)^{\frac{3}{2}}} \right) \qquad (2)$$

Or, written in the form of a series,

$$f = -\frac{v^2}{r}\left\{ \cos \theta + \cos 2\theta \left(\frac{1}{s} + \frac{1}{4s^3} + \frac{15}{128} \cdot \frac{1}{s^5} + \ldots \right) + \cos 4\theta \left(\frac{1}{4} \cdot \frac{1}{s^3} + \frac{3}{16} \cdot \frac{1}{s^5} + \ldots \right) + \&c. \right\} \qquad (3)$$

Where l = length of connecting-rod
and s = length of connecting-rod ÷ radius of crank.

As neither of these expressions are convenient to use, it is necessary to adopt the nearest approximation that will admit of a tangible interpretation.

The first approximation is $f = -\frac{v^2}{r} \cos \theta$, which is the same as (1), and is therefore the value of f to use when the connecting-rod is of infinite length, or, in other words, when its "obliquity" is neglected.

The second approximation is

$$f = -\frac{v^2}{r}\left(\cos \theta + \frac{\cos 2\theta}{s} \right) \qquad (4)$$

and this is sufficiently accurate for all ordinary calculations and makes an almost complete allowance for the obliquity of the connecting-rod, for the other terms of the series diminish very rapidly in value. This expression (4) will therefore be used in the problems of finding the actual values of the disturbing forces in an engine.

The error involved by using this expression instead of the complete series or equation (2) is, if $s = 4$, 1·56 per cent., and if $s = 5$, 0·8 per cent. This error is therefore very small indeed, and in any graphical calculations would hardly be appreciable.

Means of Balancing the Piston and Rod.—Expression (4) may be written in the form $f = -\frac{v^2}{r}\cos\theta - \frac{v^2}{rs}\cos 2\theta$. Cos θ and cos 2θ are both simple harmonic functions but with this difference, cos θ has *one* period per revolution of the engine and cos 2θ has *two* periods per revolution. This will be seen at a glance in fig. 289, where both these curves and their resultant are drawn.

Let M_1 = weight of piston, rod, and crosshead. Now, in order to balance the inertia force, $F = -\frac{M_1}{g}\cdot\frac{v^2}{r}\cos\theta - \frac{M_1}{g}\cdot\frac{v^2}{rs}\cos 2\theta$, two weights must be fitted so that one of them has an acceleration in the same line of motion as the piston, equal to $+\frac{v^2}{r}\cos\theta$, and the other has an acceleration also in the same line of motion, equal to $+\frac{v^2}{rs}\cos 2\theta$, both weights being equal to M_1.

Or, the weights W_1 and W_2 may have different values and radii of influence (r_1 and r_2), so long as

$$-\frac{M_1 v^2}{g r}\cos\theta = \frac{W_1 v_1^2}{g r_1}\cos(180^\circ + \theta),\ i.e.,\ W_1 r_1 = M_1 r$$

and

$$-\frac{M_1 v^2}{g s r}\cos 2\theta = -\frac{W_1 v_2^2}{g s r_2}\cos 2(180^\circ + \theta),\ i.e.,\ W_2 r_2 = M_1 r.$$

Considering the first force, it will be obvious that if a weight W_1 be fitted to revolve with the crank-shaft and opposite to the crank, it will have the required vertical acceleration and, therefore, the required counteracting inertia force.

This is the method that is in the majority of cases adopted.

It has one drawback, however; another inertia force is introduced in a horizontal direction due to its harmonic motion in that direction.

This force is equal to $-\frac{W_1 v_1^2}{g r_1}\sin\theta$.

The main result of this method is really to overbalance the engine in a horizontal direction, as will be seen in a subsequent paragraph.

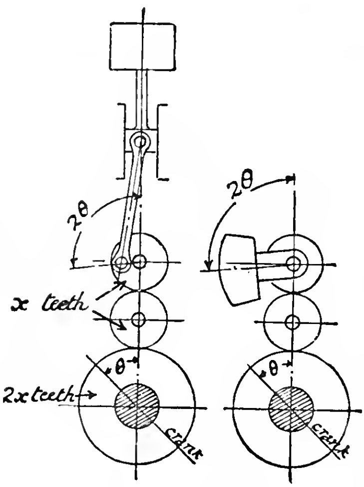
Practically, two weights would have to be fitted each equal to half W_1, and on each crank web.

A more correct method is that suggested by Mr. Yarrow to fit two (for practical convenience) "bob" weights, driven by eccentrics, keyed on the shaft. $\frac{W_1}{2}$ should equal the weight of the bob, eccentric, and rods, and the eccentrics of throw r_1 must be placed opposite to the crank. The only error involved in this method is the slight horizontal inertia force due to the weight of the eccentrics and obliquity of the eccentric-rods.

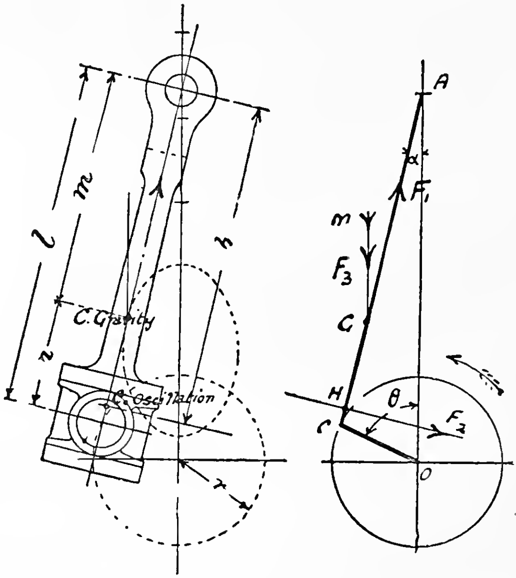
Considering the second force $-\frac{M_1 v^2}{g s r}\cos 2\theta$, a mechanical difficulty is confronted here in producing a fitting that shall move with an acceleration proportional to cos 2 (180° + θ). Two ways in which this could be done are shown in figs. 291, 291*a*. The figures are self explanatory. Another way would be by means of a cam fixed to the shaft that would cause a weight to move up and down twice per revolution and satisfy the condition that its acceleration would be always proportional to cos 2 (180° + θ).

There are possibly other mechanical devices for attaining the same object.

Now, these methods, although very well in theory, would be altogether too inconvenient for actual practice. Thus it is that this part of the disturbing force on an engine is neglected as far as actual balancing is



Figs. 291 and 291*a*.



Figs. 292 and 292*a*.

concerned. Doing so is generally termed "neglecting the obliquity of the connecting-rod."

5. **Inertia of the Connecting-rod.**—In finding the acceleration of the connecting-rod a problem of extreme complexity is presented, and it is usual and necessary, therefore, in practice to make certain approximations and assumptions in treating the subject. The connecting-rod having its top end moving in a straight line and its bottom end in a circle, has consequently a different total acceleration at all points in its length. Adopting the approximation in par. 4, equation 4, the top end has an acceleration the same as the piston-rod, which is therefore $f = -\frac{v^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{s}\right)$.

The bottom end moving with a uniform angular velocity about the axis of the shaft has a vertical acceleration $-\frac{v^2}{r}\cos\theta$ and a horizontal acceleration $\frac{v^2}{r}\sin\theta$.

Any point on the connecting-rod between the centres will have an elliptic path, the minor axis of which varies from $2r$ to O, and the vertical acceleration varies from $-\frac{v^2}{r}\cos\theta$ to $-\frac{v^2}{r}\left(\cos\theta+\frac{\cos 2\theta}{s}\right)$. If the mass of the whole rod be supposed concentrated at the centre of gravity the acceleration of the whole rod vertically will be

$$-\frac{v^2}{r}\left(\cos\theta+\frac{n}{l}\cdot\frac{\cos 2\theta}{s}\right). \qquad \text{(fig. 292)} \quad - \quad (5)$$

If it is desired to find the true value of the inertia of the rod in order to combine it with the "steam" twisting moments, or for reasons of other interest, the strict investigation is outlined in the footnote, but as far as balancing is concerned, it is out of the question for practical reasons to use such elaboration.

Considering expression (5) for the acceleration, it is obvious from the remarks on the inertia of the piston-rod (§ 4) that a balance weight can be fitted to the engine that shall have an acceleration proportional to $\cos\theta$, but that it is impossible practically to fit one having an acceleration proportional to $\cos 2\theta$. The part $-\frac{v^2}{r}\cdot\frac{n}{l}\,\frac{\cos 2\theta}{s}$ has then perforce to be neglected.*

* Let θ be angular position of the crank from the top centre and α the angle the connecting-rod makes with the centre line. Then at any instant the angular velocity of the connecting-rod is $\frac{d\alpha}{dt}$ and the angular acceleration of the connecting-rod is $\frac{d^2\alpha}{dt^2}$. There are three forces on the rod which determine its resultant acceleration (fig. 292*a*):—

$$F_1 = \frac{Mm}{g}\left(\frac{d\alpha}{dt}\right)^2 \text{ acting along the rod.}$$

$$F_2 = \frac{Mm}{g}\,\frac{d^2\alpha}{dt^2} \text{ acting at right angles to the rod through H.}$$

$$F_3 = \frac{M}{g}f \text{ acting through G parallel to the centre line.}$$

Where M = weight of the rod.
f = acceleration of rod vertically.
g = acceleration of piston-rod.

It may easily be shown that—

$$\frac{d\alpha}{dt} = \frac{v\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}$$

$$\frac{d^2\alpha}{dt^2} = -\frac{v^2(l^2 - r^2)\sin\theta}{r(l^2 - r^2\sin^2\theta)^{\frac{3}{2}}}$$

$$f = -\frac{v^2}{r}\left(\cos\theta + \frac{rl^2\cos 2\theta + r^3\sin^4\theta}{(l^2 - r^2\sin^2\theta)^{\frac{3}{2}}}\right)$$

By substitution, each of the three forces can be found. In addition to the forces there is the deadweight of the rod = M acting vertically through G. To find the effect on the twisting moment, the moments of the four forces must be taken about O for different values of θ and added algebraically to the remainder of the twisting moment diagram. The proof of the above formulæ may be found in advanced treatises on mechanics and steam.

Means of Balancing the Connecting-rod.—Neglecting, therefore, the part involving $\cos 2\theta$, the vertical acceleration of the rod is $\frac{v^2}{r}\cos\theta$. To balance by a rotating weight, a weight must be placed in the same position as that for balancing the piston, &c. If M_3 equal the weight of connecting-rod, then the weight W_3 is given by $W_3 r = M_3 r$. This weight will partly balance the rod in a horizontal direction. It will, however, really overbalance horizontally, because, although the whole rod has a vertical acceleration $\frac{v^2}{r}\cos\theta$, only part of it has an acceleration $\frac{v^2}{r}\sin\theta$ (which has the same maximum value or amplitude as $\frac{v^2}{r}\cos\theta$, its phase being 90° behind). Therefore, if it is desired to balance any particular engine horizontally in preference to vertically, the full weight W_3 must not be put on, but only a portion of it, which for a fairly accurate approximation may be such that $\frac{m}{l} M_3 r = W_3 r$. (fig. 292).

This is one of the assumptions alluded to at the beginning of the paragraph, and really amounts to dividing the connecting-rod into two parts, and assuming the part $\frac{n}{l} M_3$ to be concentrated at the crosshead and $\frac{m}{l} M_3$ to be concentrated at the crank-pin.

Professor Dalby gives a division of the rod that is said more accurately to represent the case, the division being as follows:—At the crank pin $M_3\left(\frac{m \times h}{l^2}\right)$; the remainder at the crosshead, h, being the distance from crosshead centre to centre of oscillation.

This necessitates an experiment with one of the actual rods to find the centre of oscillation.

The finding of the centre of gravity does not require any special experiment, for in ordinary shop transit the slings are usually placed in this position. In case that either observations are not readily available, the following table of typical connecting rods is given :—

(*v.* Fig. 296.)	Connecting-Rod to		
	Fig. 52.	Fig. 58.	Fig. **297.**
Position of centre of gravity from top centre,	·675 l	·58 l	·643 l
,, ,, oscillation ,, ,,	·88 l	·912 l	·99 l
$\frac{m \times h}{l^2}$,	·594	·528	·637

If it is desired to balance the engine as far as possible, both vertically and horizontally, then the reciprocating part of the connecting-rod must be balanced by means of reciprocating bob weights as explained and the rotating part must be balanced by a rotating weight.

6. **Inertia of Crank.**—The balancing of the crank is the simplest part of the whole problem. The unbalanced part, of course, consists of the

crank webs beyond the shaft and the crank-pin. The acceleration of the centre of gravity vertically is $f = -\frac{v^2}{r}\cos\theta$, and horizontally $f = \frac{v^2}{r}\sin\theta$, or, if referred to a horizontal axis, the horizontal acceleration is $f = -\frac{v^2}{r}\cos\theta$. Thus to balance both vertically and horizontally a weight must be placed opposite the crank-pin acting at radius r_1 such that its weight is given by $W_4 r_1 = M_4 R$, where R is the distance of the centre of gravity of the crank webs &c., from the shaft centre.

7. **Inertia of Valve Gear.**—The simple valve gear is, of course, a slider-crank chain like the piston, connecting-rod, and crank, and its treatment is therefore precisely the same.

Practically it is not convenient to fit special balance weights for the valve gear, and they are generally embodied in those for the main parts of engine. As this method introduces a disturbing couple on the engine which must be eliminated, the problem of combination will be treated at length in par. 9.

Again, most marine engines are fitted with Stephenson's link motion. It is impossible to use the formula for the acceleration of the valve in any method of balancing; in fact, the solution of the displacement, velocity, and acceleration of the slide valve is an extraordinarily complicated operation, especially if slot links are fitted.

It is usual, therefore, to adopt the following assumptions in dealing with this part of the engine :—

1. *Reciprocating Masses.*

Ahead, . .	Valve. Spindle. Half the link. Link block. 0·5 eccentric-rod if notch up gear; or 0·4 eccentric-rod if direct gear. Half the bridle-rods = 1 bridle-rod if connected to "ahead" end.
Astern, . .	Half the link. 0·5 eccentric-rod if notch up gear; or 0·4 eccentric-rod if direct gear. 1 bridle-rod if connected to "astern" end.

2. *Rotating Masses.*

Ahead, . .	Remainder of eccentric-rod. Eccentric sheave. Eccentric strap.
Astern, . .	Remainder of eccentric-rod. Eccentric sheave. Eccentric strap.

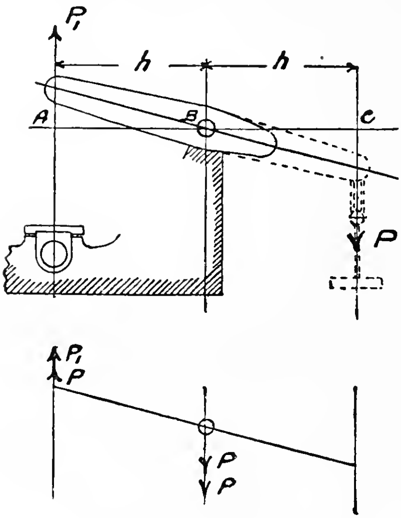
8. **Inertia of Air Pump.**—The air pump when out of centre with the engines, and having an opposite motion to that of the particular engine that drives it, presents a rather peculiar problem when its effect on the engine support or foundation is considered conjointly with the means of balancing.

Let W_5 W_6 be the equivalent weights of parts when having an acceleration f equal to that of the piston-rod (fig. 293). Then A B = B C.

Let the inertia forces due to W_5, W_6 be P, P_1 respectively $\left(P = W_5 \frac{f}{g}\right.$ and $\left.P_1 = W_6 \frac{f}{g}\right)$, and suppose them to act as shown in fig. 293 when the lever is in that position.

The force P is equivalent to a transverse couple $2\,Ph$ on the engine and a downward force P acting through the rocking shaft bearing (fig. 293*a*). There is thus a total force $P + P_1$ acting upward with the piston-rod inertia force and a force 2 P acting downward through the bearing.

Means of Balancing.—There is no practicable method of balancing the transverse couple, except perhaps by fitting an equal air pump on the opposite side of the engine and driven from the same crosshead. This would, however, be an extremely awkward fitting.



Figs. 293 and 293*a*.

Now, if h is an appreciable dimension compared with the breadth of the ship, the effect will be to produce torsional vibrations of the ship; but if h be very small compared with the same, there is on practically the same centre line a downward force 2 P and an upward force $P + P_1$. The difference of these forces is the force causing vibrations of the same kind as those produced by the engine. In practice it will be found that W_5 and W_6 are very nearly equal to one another, and in that case the two inertia forces neutralise each other, and the air pump may consequently be neglected.

In the greater number of fast-running engines h will, however, be of appreciable magnitude, and it is, therefore, usual to make the addition $W_5 + W_6$ to the engine reciprocating weights.

Note.—When in the remainder of this chapter the term "perfectly balanced" is used without comment, it may be taken as meaning that the parts under consideration are perfectly balanced as far as the limits just discussed enable them to be.

9. Single-crank Engine and Valve Gear balanced by Rotating Weights only.—Reduce all weights of moving parts to equivalent weights driven by cranks of the same radius, say the crank radius r.

Let M_1 = weight of engine reciprocating parts at radius r.
M_2 = " " rotating " "
M_3 = " valve reciprocating " "
M_4 = " " rotating " "

Imagine a plane O X Y (fig. 294) through *any* point O on the crank shaft and at right angles to it. This plane will be called the reference plane.

Fig. 294.

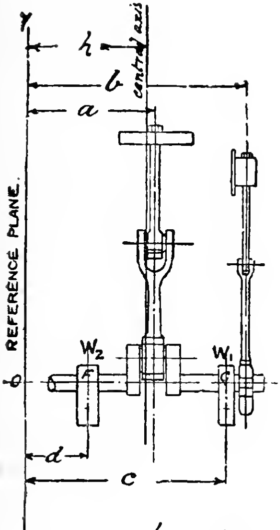


Fig. 294a.

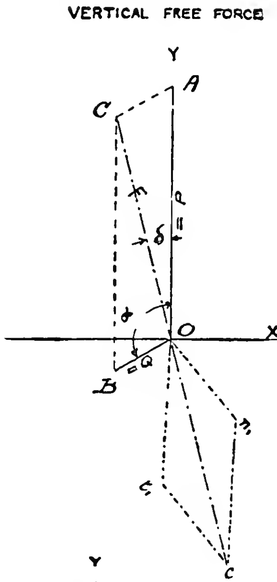
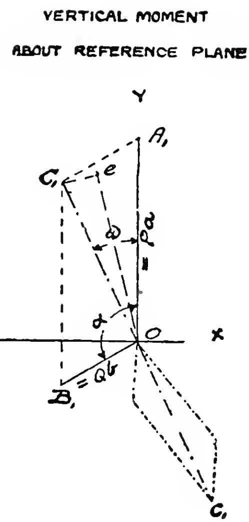


Fig. 294b.



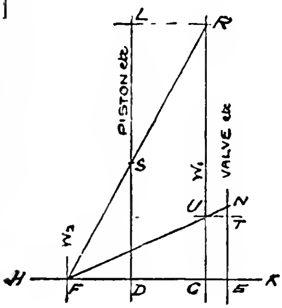


Fig. 294c.

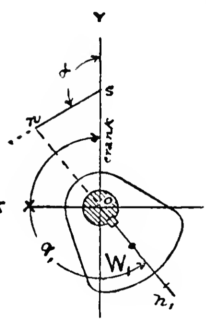


Fig. 294d.

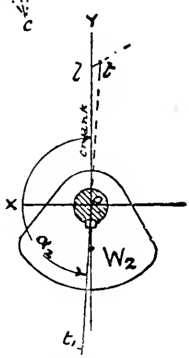


Fig. 294e.

Case 1.—Suppose it is desired to balance the engine as completely as possible in a *vertical* direction in preference to the horizontal direction.

By the theory of couples any number of coplanar forces can be reduced to a single force acting at any point accompanied by a couple.

In the engine under consideration there is at any time a free vertical force $-\frac{v^2}{gr}(M_1 + M_2)\cos\theta - \frac{v^2}{gr}(M_3 + M_4)\cos(\theta + \alpha)$, and a couple referred to the reference plane

$$= -\frac{v^2}{gr}(M_1 + M_2)\,a\cos\theta - \frac{v^2}{gr}(M_3 + M_4)\,b\cos(\theta + \alpha)$$

As the force and couple cannot be balanced by a single force, because they are not in the same phase, two weights at least must be employed to balance the engine. A couple as well as a force can be represented by a straight line of length proportional to its moment. It is usual in theoretical mechanics for this line, which is called the axis of the couple, to be drawn anywhere at right angles to the plane of the couple, but in order not to confuse the subject it will be found much more convenient to draw this line in the plane of the couple.

Project the lines of cranks on to the reference plane (shown in front elevation on figs. 294*a* and 294*b*), and measure off parts O A and O B equal to $(M_1 + M_2)$ and $(M_3 + M_4)$ respectively. Measure off O A_1 and OB_1, equal to $(M_1 + M_2)\ a$ and $(M_3 + M_4)\ b$ respectively. Each of these quantities multiplied by $-\frac{v^2}{g\,r}$, of course, gives the maximum inertia forces and couples due to them, and the angle α is also the phase of those forces and couples of the crank behind those of the valve gear. $-\frac{v^2}{gr}$ being common to all, balance weights included, need not therefore enter into any calculations. Then A, B, A_1 and B_1, are the generating points of simple harmonic motions along O X, O Y. Compound these by the problem in par. 3. Their resultants are C and C_1 respectively.

Suppose it is convenient to place the rotating balance weights at points G and F on the shaft. Then, stated definitely, the problem is to find the values of the weights W_1 W_2, and the angles at which they are to be fixed relatively to the crank such that the resultant of the vertical free force produced by their rotation is equal and opposite to O C, and that the resultant of their moments about the reference plane is equal and opposite to O C_1.

For convenience let $M_1 + M_2 = P$, and $M_3 + M_4 = Q$. Let the angles W_1 W_2 make with the crank be α_1 α_2 respectively.

Then these values may be found by solving the equations—

$$P + Q \cos \alpha = -(W_1 \cos \alpha_1 + W_2 \cos \alpha_2) \qquad (1)$$

$$Pa + Q\, b \cos \alpha = -(W_1 c \cos \alpha_1 + W_2 d \cos \alpha_2) \qquad (2)$$

$$W_1^2 (c - d)^2 = P^2 (a - d)^2 + Q^2 (b - d)^2 - 2\,P\,Q\,(a - d)(b - d) \cos \alpha \quad (3)$$

$$W_2^2 (c - d)^2 = P^2 (c - a)^2 + Q^2 (b - c)^2 - 2\,P\,Q\,(c - a)(b - c) \cos \alpha \quad (4)$$

Care must be taken that the distances are positive or negative according to their sense. These are very easily solved if the known values of the constants be substituted immediately.

The *graphical solution*, however, is more quickly arrived at.

Draw a base line H K, and a pair of axes O X, O Y (figs. 294*c*, 294*d*, and 294*e*). On H K mark off the engine centres, &c., to scale, and erect perpendiculars as shown.

On either of the balance weight centres (say that of W_1) mark off G R = P, and G U = Q.

Draw horizontals through R and U. Join R F, cutting D L in S. Join U F, cutting E T in N. Then, by the principle of the "lever," the part of D L or P that must be apportioned to W_1 is D S, and the part to W_2 is S L.

Similarly, the part of E T to be apportioned to W_1 is E N and the part to W_2 is T N. E N is necessarily greater than E T, because the valve centre is outside both the centre of W_1 and W_2.

To find W_1 mark off (fig. 211*d*) os = D S and sn = E N. Join no and produce. Then $on = W_1$ and on_1, its direction relatively to the crank line oy.

To find W_2 mark off ol = S L (fig. 294*e*), and lt = − T N. T N being negative the direction must be in the opposite direction to sn.

Then $W_2 = ot$ and ot is its position relatively to the crank line O Y.

If the work has been done correctly, the resultant of the two weights W_1 W_2 will be found to be equal and opposite to O C, and the resultant of their moments about the arbitrary reference plane will be found to be equal and opposite O C_1 (figs. 294*a* and 294*b*).

The centre of gravity of W_1 and W_2 must be at radius r, but, if necessary for it to be at radius r_1, then $W_1r = w_1r_1$, and $W_2r = w_2r_2$. Thus the engine is balanced perfectly in a vertical direction for both the distributing couple and the free up and down force.

Case 2.—Suppose it is desired to balance the engine as completely as possible in a *horizontal* direction in preference to the vertical direction.

Here the rotating parts of the engine have simply to be dealt with, because the reciprocating parts have no resolved part horizontally.

Apply the same process as for Case 1, but instead of using $M_1 + M_2$ and $M_3 + M_4$ use M_2 and M_4 respectively.

It will now be seen, as intimated in the latter part of par. 5, that the engine is over-balanced horizontally when completely balanced vertically by rotating balance weights.

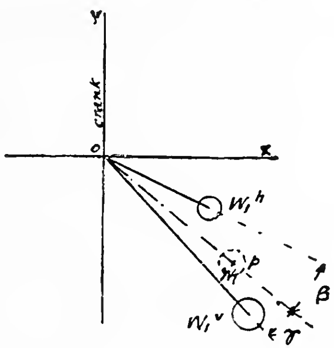


Fig. 295.

Case 3.—Suppose it is desired to balance the engine as completely as possible in both directions, the balance weights being rotary only.

The only thing that can be done here is to effect a compromise.

Find, as in Cases 1 and 2, the weights necessary for balancing in both directions separately. Let these weights be W_1^v and W_2^v for the vertical balance and W_1^h and W_2^h for the horizontal balance. Draw their centre lines as in fig. 295. They will most probably not be on the same centre line. Divide the angle between them so that $\frac{\gamma}{\beta} = \frac{W_1^h}{W_1^v + W_1^h}$.

Then Yop is the best angle for the weight to be fitted. Also make $op = \frac{W_1^v + W_1^h}{2}$. Then op is the value of the weight W_1 to be fitted.

Treat W_2^v and W_2^h in a similar manner.

The engine is thus incompletely, and at the same time as completely, balanced as possible under the condition of having only rotating balance-weights.

10. Single-crank Engine and Valve Gear Balanced by Bob Weights and Rotating Weights combined.—The engine may be balanced perfectly both vertically and horizontally by the above combination.

In order to do so the bob weights are made to balance the reciprocating parts, and the rotating weights are made to balance the rotary parts.

To find the bob weights proceed exactly as before, using M_1 and M_3 for P and Q. The weights found are those which would be driven by eccentrics of throw r. This radius will in most cases be too large for convenience. If r_1 be the convenient throw, then $W_1 r = B_1 r_1$, and $W_2 r = B_2 r_1$, B_1 and B_2 being the new weights which should include the weight of the eccentric and rod. To find the rotating balance weights again proceed as before, using M_2 and M_4 for P and Q respectively, but also taking into account the weight of the bob weight eccentrics, as they introduce a horizontal disturbance which must be eliminated.

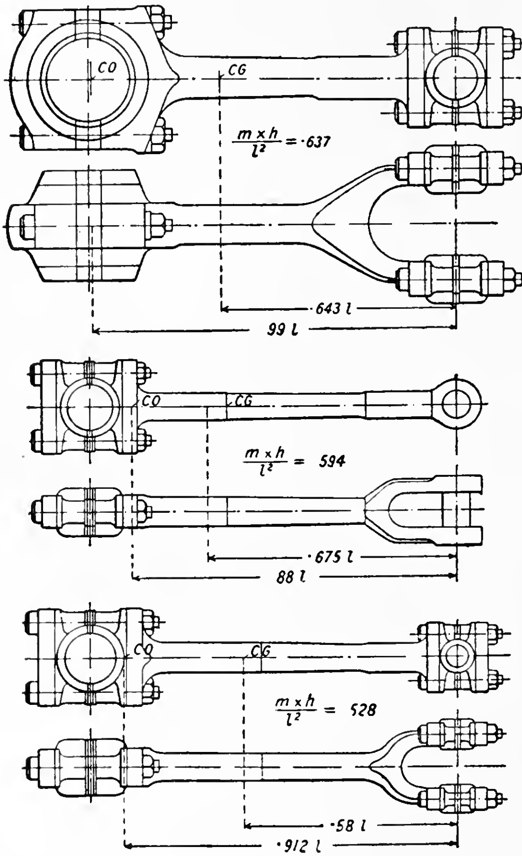


Fig. 296.

The solution of the problem of balancing by this particular method, of course, depends on the fact that the bob weights have no resolved part horizontally.

11. To Balance an Engine of any number of Cranks.—As the method of solving this problem is the same in principle as has just been described in pars. 9 and 10, the case will be most easily explained by means of an example, having particular reference to the three-crank engine.

In the example the engine will be balanced both vertically and horizontally, and the combined system of bob and rotating weights must, therefore, be used. Should, however, the designer of any particular engine decide that he cannot very well adopt the most complete method, he will understand from the previous paragraphs what sacrifices will have to be made if only one kind of balancing device is used. The methods of construction are precisely the same, although more extensive.

The example (fig. 297) is a three-cylinder engine with ordinary link motion for each valve. Two bob weights will be fitted, one at each end of the engine, and four balance weights, one on each crank web of the outside engines. It will also be decided to balance half the rotating masses by the H.P. and L.P. forward weights, and half by the H.P. and L.P. after weights.

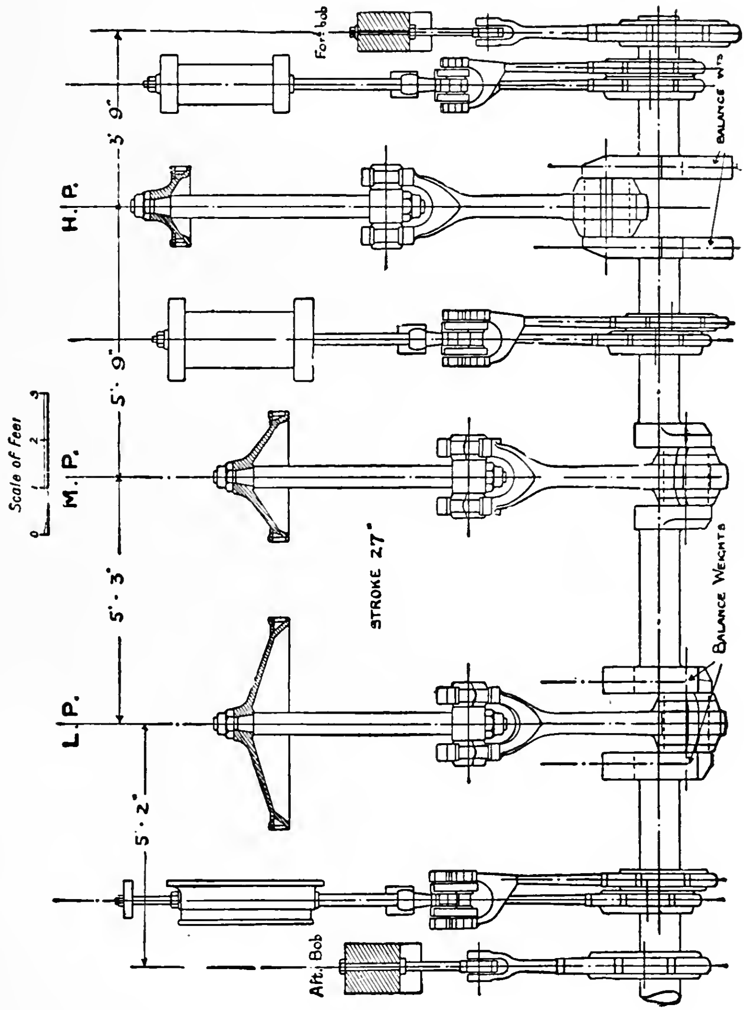


Fig. 297.

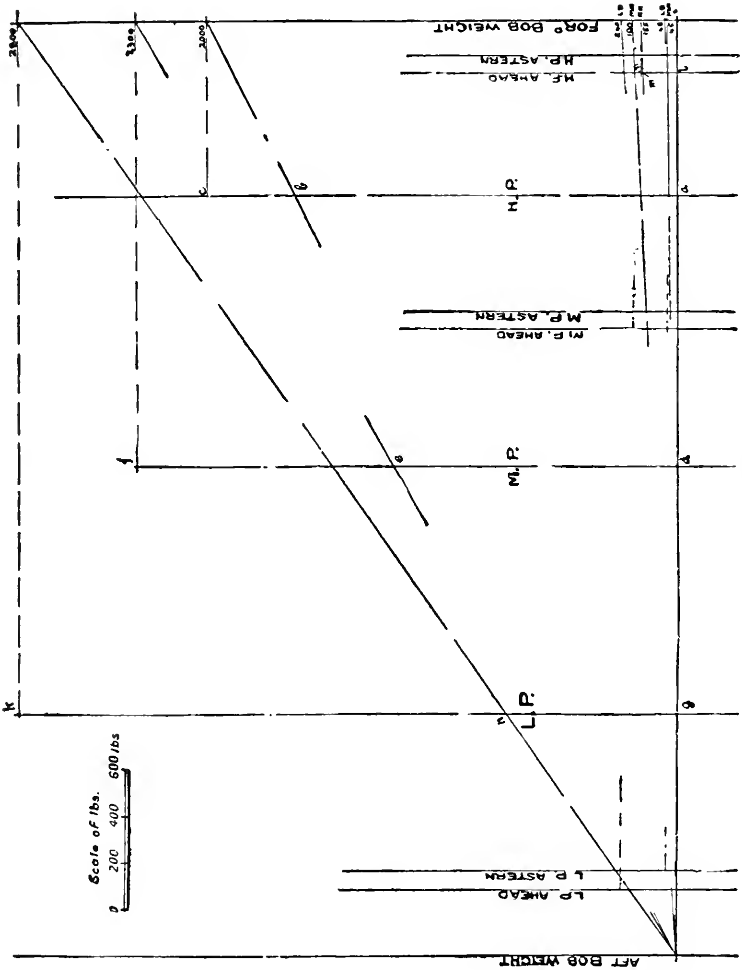

FOR° BOB WEIGHT
H.P. ASTERN
H.P. AHEAD
H.P.
M.P. ASTERN
M.P. AHEAD
M.P.
L.P.
L.P. ASTERN
L.P. AHEAD
AFT BOB WEIGHT
Scale of lbs.
0
200
400
600 lbs

Fig. 298.

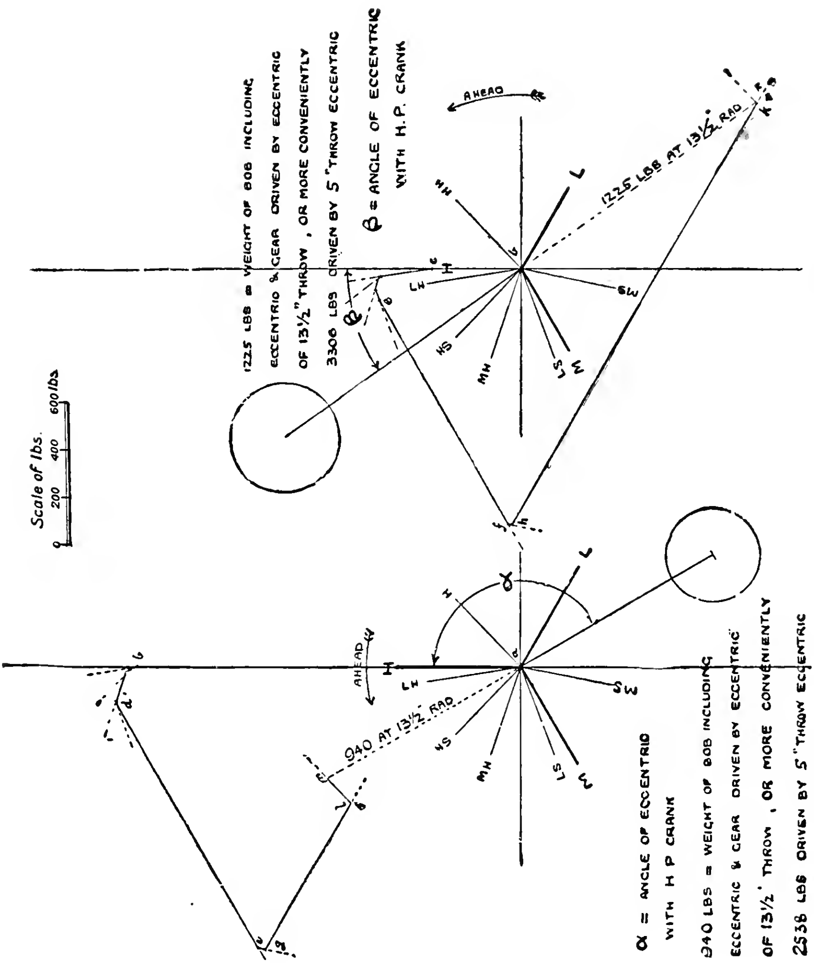


Fig. 298b.

Fig. 298a.

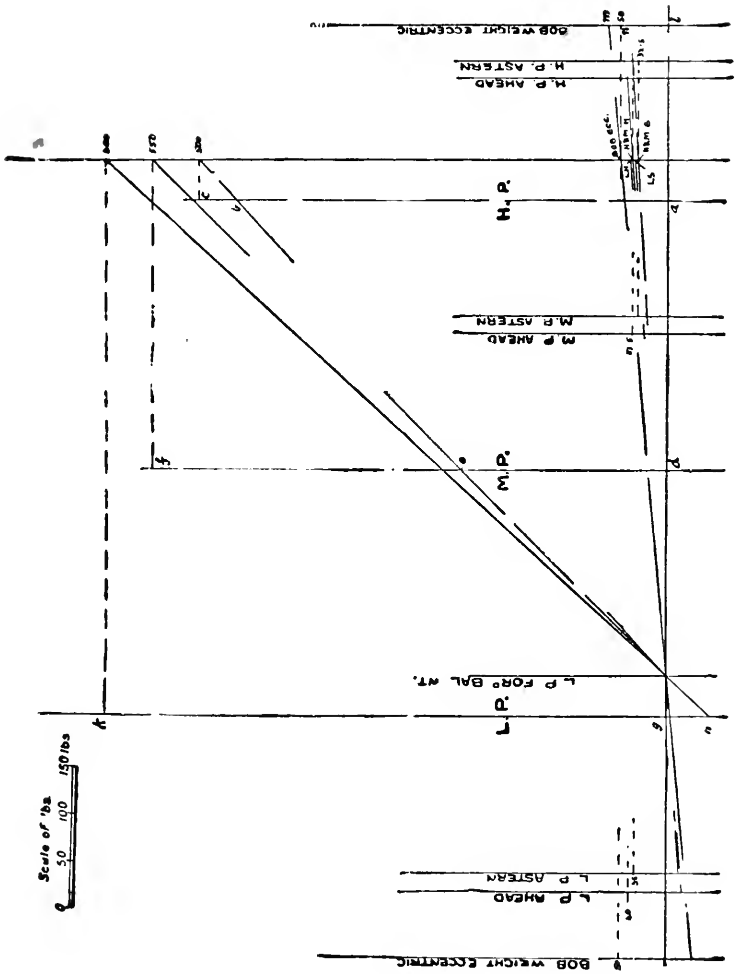

H.P.
M.P.
L.P.
H.P. AHEAD
H.P. ASTERN
M.P. AHEAD
M.P. ASTERN
L.P. AHEAD
L.P. ASTERN
BOB WEIGHT ECCENTRIC
BOB WEIGHT ECCENTRIC
L.P. FOR° BAL. WT.
Scale of lbs
0
50
100
150 lbs

Fig. 299.

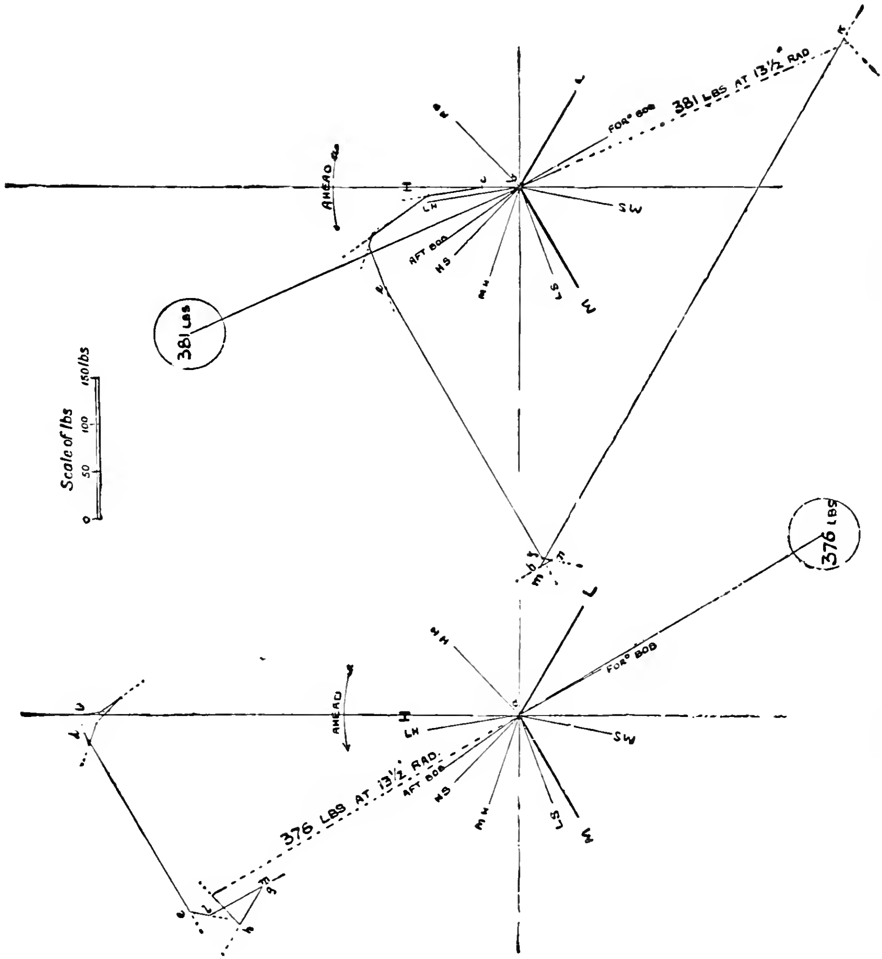


Fig. 299b.

Fig. 299a.

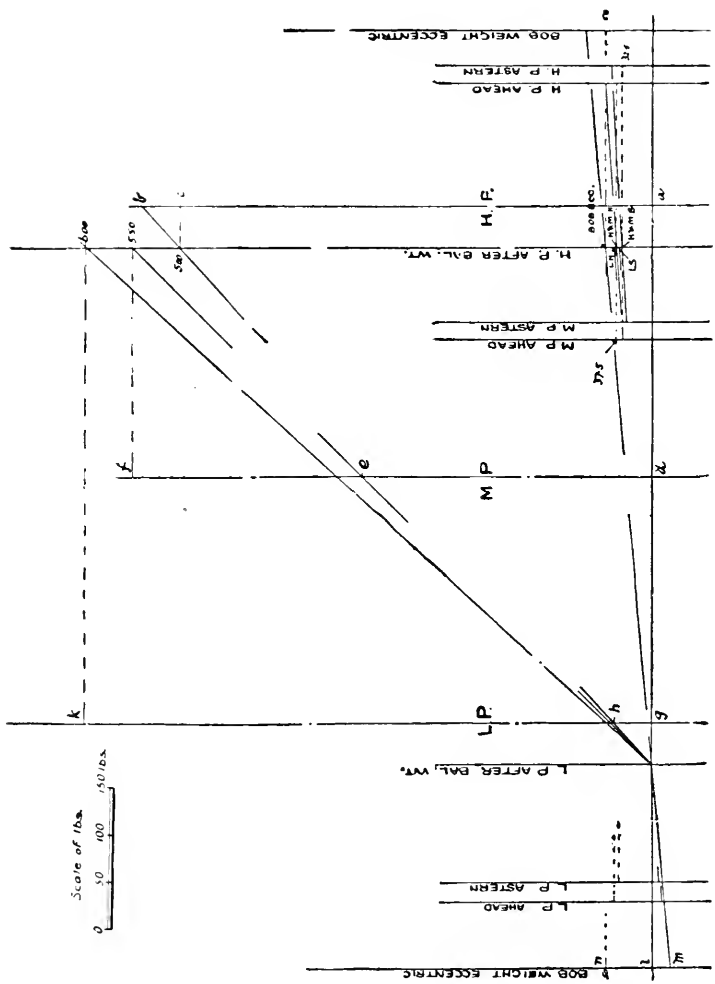

BOB WEIGHT ECCENTRIC
H. P. ASTERN
H. P. AHEAD
H.P.
H. P. AFTER BAL. WT.
M P ASTERN
M P AHEAD
M P
L.P.
L P AFTER BAL. WT.
L P ASTERN
L P AHEAD
BOB WEIGHT ECCENTRIC
600
550
500
375
Scale of lbs.
0
50
100
150 lbs.

Fig. 300.

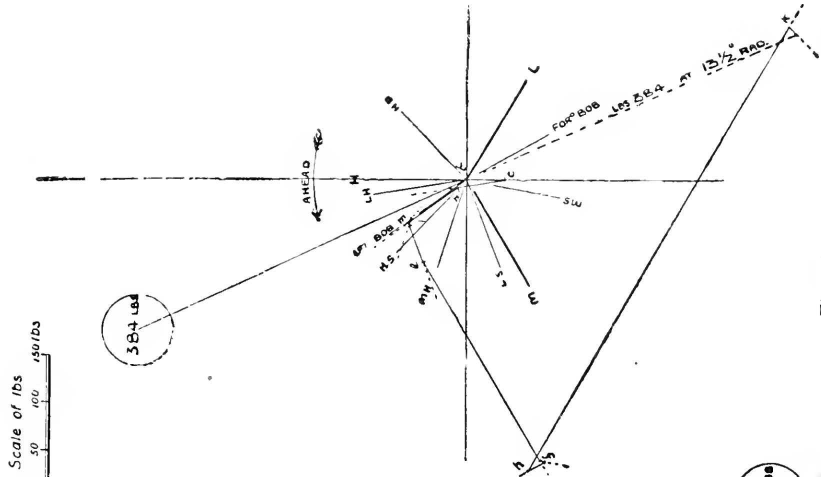


Fig. 300b.

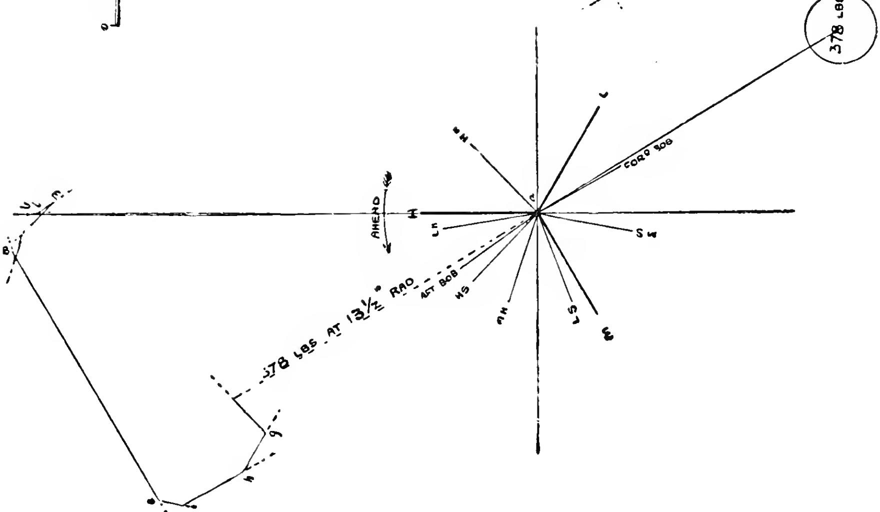


Fig. 300a.

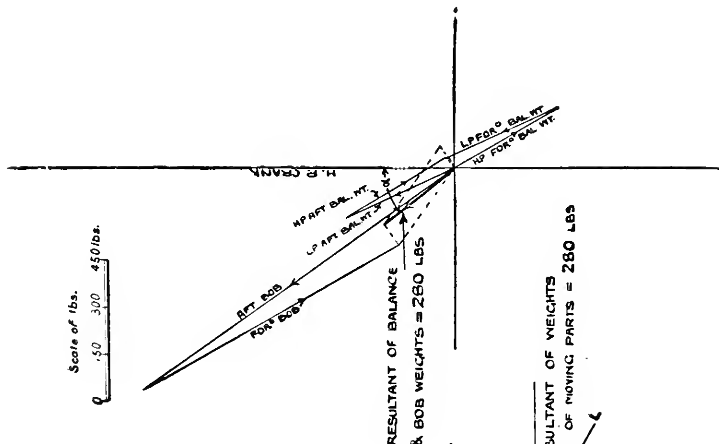


Fig. 302.

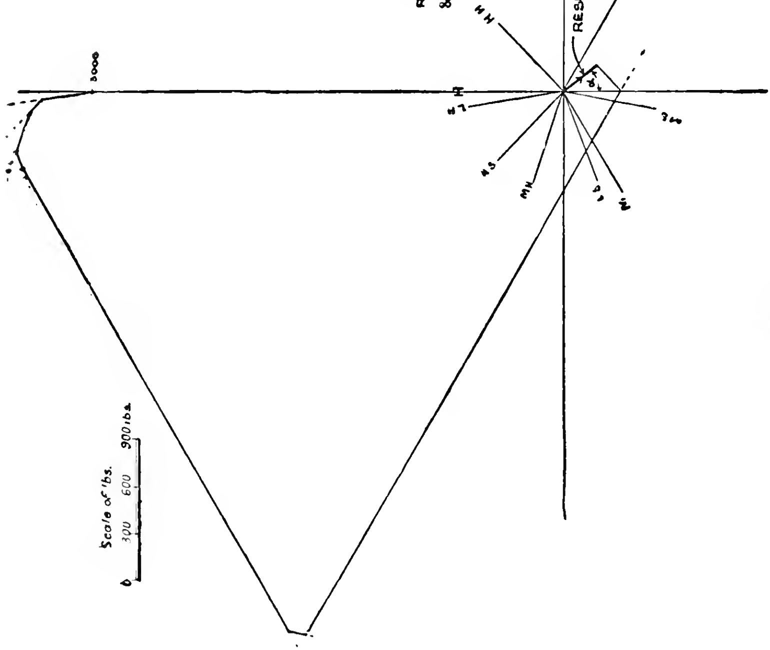


Fig. 301.

The weights of the moving parts are tabulated as follows, and the diagrams are drawn in figs. 298 to 302.

TABLE XCI.—BALANCING ENGINES: WEIGHTS OF MOVING PARTS.

	Reciprocating Weights.			Rotating Weights.			Half Rotating Weight.	Sum of Reciprocating and Rotating Weights.
	Actual Weight of Reciprocating Parts.		Equivalent Weight at Radius of Crank, 13½ Inches.	Actual Weight of Rotating Parts.		Equivalent Weight at Radius of Crank, 13½ Inches.		
	Lbs.	At Radius, Ins.	Lbs.	Lbs.	At Radius, Ins.	Lbs.	Lbs.	Lbs.
H.P. engine, .	2,000	13½	2,000	1,000	13½	1,000	500	3,000
,, ahead, .	698	3	155	338	3	75	37½	230
,, astern, .	202	3	45	292	3	65	32½	110
M.P. engine, .	2,300	13½	2,300	1,100	13½	1,100	550	3,400
,, ahead, .	855	3	190	338	3	75	37½	265
,, astern, .	202	3	45	292	3	65	32½	110
L.P. engine, .	2,800	13½	2,800	1,200	13½	1,200	600	4,000
,, ahead, .	1,036	3⅛	240	345	3⅛	80	40	320
,, astern, .	207	3⅛	48	302	3⅛	70	35	118
Forward bob,	2,538	5	940	Eccentric.		100	50	...
After bob, . .	3,306	5	1,225	,,		100	50	...
H.P. forward,	Balance Weights.			376	13½	...	...	...
,, after, .				378	13½	...	...	..
L.P. forward,				381	13½	...	..	...
,, after, .				384	13½	...	...	...

Fig. 298, 298*a*, 298*b* are for finding the bob weights;

Fig. 299, 299*a*, 299*b* for the H.P. and L.P. forward balance weights.

Fig. 300, 300*a*, 300*b* for the H.P. and L.P. after balance weights.

Fig. 301 finds the resultant of the vertical free force by using the sum of the reciprocating and rotating parts.

Fig. 302 finds the resultant of the vertical free force of all the bob and balance weights.

If the work is done correctly these two resultants will be found to be equal and opposite to each other.

The figures and construction will be found self-explanatory after reading pars. 9 and 10.

12. So far, it has been seen that, in order to balance an engine, it is only necessary to know the relative values of the inertia forces and not the actual value. But, in order to compare one design of an engine with another, it is necessary to know the actual values (or at least relative values referred to a common basis) of the resultant disturbing forces and couples.

For instance, in comparing submitted designs of engines for any particular ship, it is obvious that, in a general sense, the less the resultant forces and couples are, the less the vibration, and, therefore, the better the engine.

Referring to par. 4, *et seq.*, dealing separately with the inertia of the various parts, and having balanced the engine as perfectly as practically possible, it will be seen that the disturbing forces left unbalanced are :—

For the piston, rod, and crosshead a vertical force $-\dfrac{M_1 v^2}{g r} \dfrac{\cos 2\theta}{s}$.

For the connecting-rod a vertical force $-\dfrac{M_3 v^2}{g r} \dfrac{n}{l} \dfrac{\cos 2\theta}{s}$.

For the valve gear there is a very minute quantity, which is a function of 2θ and the higher multiples of θ, and which may very reasonably be neglected.

The residual resultants in the cases of the single cylinder engine will now be considered, and the methods and principles of constructing the various resultants will apply equally well to a multi-cylinder engine.

13. *Case* 1 (corresponding to Case 1, p. 553).—The residual vertical inertia force is $x \cos 2\theta$, where $x = {}^*\left(M_1 + \frac{n}{l} \text{ connecting-rod}\right)\frac{v^2}{g\,rs}$ and is constant. There is no residual couple, seeing that the very small force left from the valve gear is neglected.

Put $\theta = 0$, then $\cos 2\theta = 1$ and x is the amplitude, par. 2, of the force, and is the quantity that can be used for comparative purposes.

To draw the curve $x \cos 2\theta$, the ordinates for various values of θ are easily determined graphically. Draw a quadrant of a circle of radius x to any suitable scale (fig. 303). Divide the arc into a convenient number of parts and drop perpendiculars on to the axis O X as shown. Draw a base line A B and divide it into *twice* the number of parts into which the complete

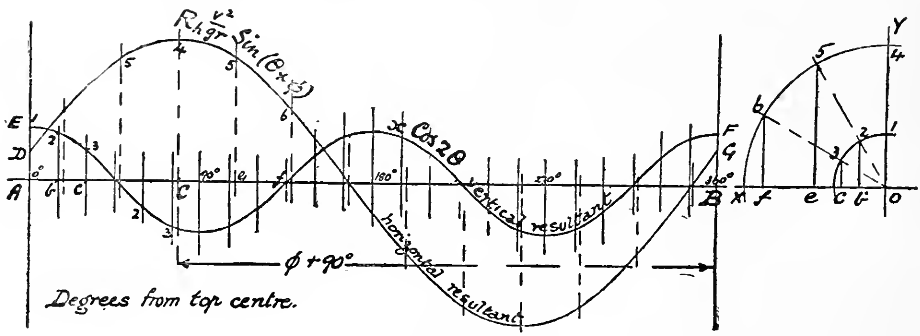


Fig. 303.

circle is divided, because the curve has two periods per revolution. Mark the divisions and degrees and erect perpendiculars to A B through the division points as shown. Mark off A 1 = O 1, b 2 = b 2, c 3 = c 3, and so on. Then the curve through E, 3, F is the required curve, the ordinates of which give the residual resultant disturbing or unbalanced inertia force at any position θ of the crank from the top centre. The curve of resultant moment about any reference plane may be drawn in a similar manner—for instance, that plane through a node of the ship. As in this case the engine is overbalanced horizontally there is a residual horizontal disturbing force.

Let the resultant of the balance weights W_1 and W_2 be W_v (fig. 304). Also, let the resultant of the rotating parts of the engine be R, both these resultants being drawn in their proper relation to the crank. Find the resultant R_h of W_v amd R. It makes an angle ϕ with the crank—that is to say, its phase is $\phi°$ before that of the crank (pars. 2 and 3).

Then the resultant horizontal force is $R_h \frac{v^2}{g\,r} \sin(\theta + \phi)$.

* In the following pages the inclusive minus signs are ignored.

Since the resultant is φ° in front of the crank, and also since the axis of reference is 90° in front of the vertical axis, the ordinate of maximum amplitude is $(\varphi + 90)^\circ$ in front of the ordinate B, and is, therefore, drawn through the point C. Divide the base line off again as before, but the number of divisions is the same as that of the complete circle, as there is here only one period per revolution. Draw the circle 4, 5, 6, and set off the ordinates as before. Then the curve D_4 G (fig. 303) is the required curve, the ordinates of which give the residual resultant horizontal unbalanced force at any position θ of the crank from the top centre.

Case 2.—Engine balanced perfectly horizontally. Here there is no residual horizontal force, and the curve therefore coincides with the base line, the amplitude always being zero.

In a vertical direction there is still the residual force $x \cos 2\theta$ as in case 1, but now there is an additional force due to the difference between the resultant of the weights required for vertical balancing and those that have been fitted for horizontal balancing.

Let W_1 W_2 be the weights as fitted, and W_h their resultant (fig. 305).

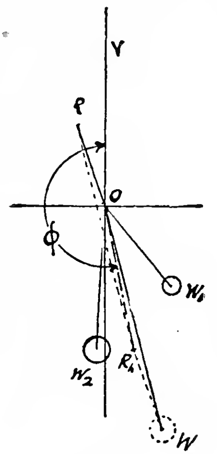


Fig. 304.

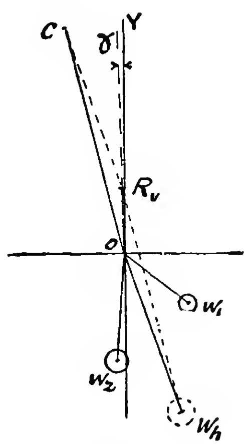


Fig. 305.

O C is the resultant of the reciprocating parts of the engine as drawn in fig. 294*a* (function of θ). Find the resultant of O C and $W_h = R_v$.

Then the resultant inertia force vertically is $R_v \frac{v^2}{g\,r} \cos(\theta + \gamma) + x \cos 2\theta$.

Although in Case 1 it was not absolutely necessary to draw the curve of force for comparative purposes, it is necessary here, because the maximum amplitude of the resultant is not the sum of the two amplitudes in the above expression.

Draw the curve $R_v \frac{v^2}{g\,r} \cos(\theta + \gamma)$, noting that its phase is γ° in front of the curve $x \cos 2\theta$.

Draw the curve $x \cos 2\theta$.

The resultant is the sum of the ordinates.

These curves are drawn in fig. 306.

The total amplitude H K is that required for comparative purposes.

This may seem on first reading a rather laborious process, but if the principles be thoroughly understood it will be found to be many times more rapid than drawing the curves by calculating the ordinates for various positions of the crank.

Case 3.—Engine balanced completely horizontally and vertically.

Here the residual horizontal force is zero, and the residual vertical force is the same as that in Case 1.

The curves just drawn are for a velocity, v, of the crank pin, corresponding to a certain number of revolutions, N, per minute.

The amplitude for any other number of revolutions, N_1, will be that just found multiplied by $\left(\frac{N_1}{N}\right)^2$.

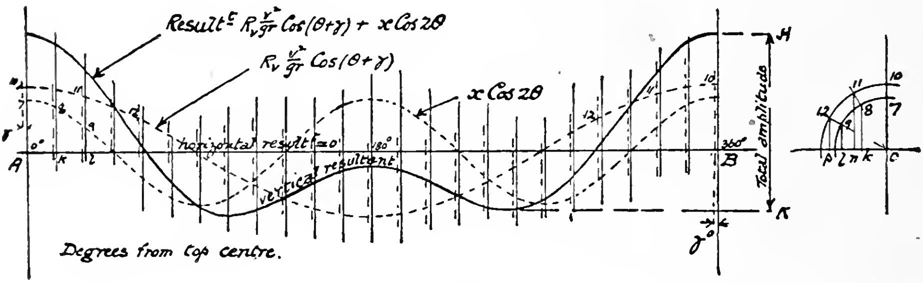


Fig. 306.

14. Curves of Free Force and Couples for a Multi-Crank Engine.—This will, as in the case of balancing (par. 11), be illustrated by means of the example of a three-cylinder engine—

Let M_1 = weight of H.P. reciprocating parts.
M_2 = ,, M.P. ,, ,,
M_3 = ,, L.P. ,, ,,

1. *Free Force.*—As the engine is as perfectly balanced as possible in both vertical and horizontal directions, the vertical free force will only be the resultant of the three residual forces $M_1 \frac{v^2}{gr\,s} \cos 2\,\theta$, $M_2 \frac{v^2}{gr\,s} \cos 2\,(\theta + 120°)$, $M_3 \frac{v^2}{gr\,s} \cos 2\,(\theta + 240°)$.

For convenience let the constants $M_1 \frac{v^2}{gr\,s}$, $M_2 \frac{v^2}{gr\,s}$, $M_3 \frac{v^2}{gr\,s} = x, y, z$ respectively.

Draw axes O X, O Y (fig. 307) and let the H.P. crank be along O Y.

In finding the resultant of the above three inertia forces or harmonic functions, *note particularly* that the angles between the generating lines are not the same as the angles between the cranks, because the forces are functions of $2\,\theta$. From A to B (fig. 308), which is 180°, really represents 360°, as far as the curves are concerned, and the sequence of the generating lines is as if the H.P. were leading instead of the L.P.

Then in fig. 307 the generating lines are in the order as shown. Find the resultant O R by the polygon as shown (par 3).

Then the resultant virtually leads the H.P. generating line by $\alpha°$, or when referred back to the real degree scale (fig. 308) by $\frac{\alpha°}{2}$.

The ordinate of maximum amplitude of the resultant therefore passes through the point C, and not through the point D, where it would if the generating lines were in the same sequence as the cranks.

Draw a quadrant of a circle of radius o R and set off the resultant curve as previously explained.

The engine being balanced perfectly horizontally the amplitude of the resultant force is zero, and the curve coincides with the base line.

If the engine were not completely balanced horizontally, the residual resultant horizontal force would, of course, be a function of θ, and the generating lines would have the same sequence as the cranks.

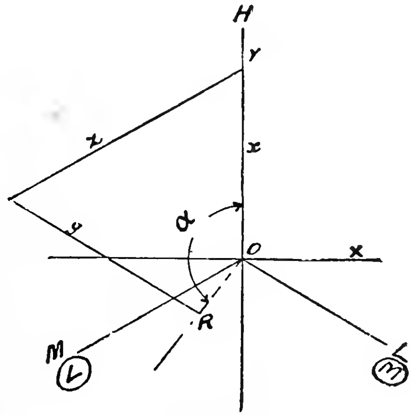


Fig. 307.

2. *Couples.*—As there are three centre lines, and a residual vertical force along each, there will be a residual resultant couple on the engine. The moment of the couple will, of course, vary according to the position of the reference plane, and in order to compare one design of an engine with another some common method of locating the reference plane is necessary.

The Admiralty, in their recent specifications, require it to be through a point on the shaft midway between the extreme cranks, and, considered

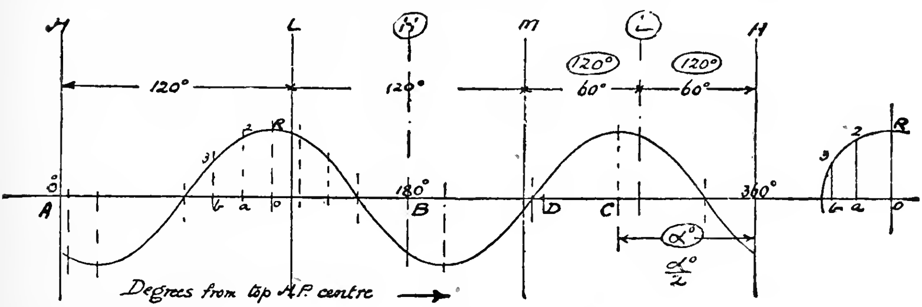


Fig. 308.

generally, this is a very reasonable point to select, because it makes the length of the engine have a minimum influence on the value of the couple, and also causes the plane to be at about the same position in the ship, whatever the design of the engine may be. This point is thus suited for marine engines, but for land engines it is advisable to select a point that will enable a stricter comparison to be made. This point should be at what is termed in theoretical mechanics "the central axis" of the couple. To explain this, let F = O C be the resultant force (figs. 294*a* and 294*b*), and O C_1 the resultant couple referred to any arbitrary plane O Y. Resolve the couple in and at right angles to the direction of the force F.

Replace the couple O*e*—that is, in the same phase plane as the force—by another equal couple but having each of its forces equal to F. Thus F h = O*e* (figs. 294*a* and 294*b*). Then F and - F on the reference plane cancel each other, leaving a force F on the new reference plane distant h from the original one and a couple $e\,C_1$ having its phase at right angles to the force and to the original couple.

Taking the case of the engine in par. 9, let the weights P = 8, Q = 2. Also, let $a = 4$, $b = 7$, and α as drawn.

Then the resultant force $F = 7{\cdot}2\ x \cos(\theta + \delta)$
and the resultant couple $O\,C_1 = 27{\cdot}8\ x \cos(\theta + \omega)$.

Where $x = \dfrac{v^2}{g\,r}$ and neglecting the obliquity of the connecting-rod.

The resolutes O*e* and $e\,C_1$ are $27{\cdot}2\ x \cos(\theta + \delta)$
and $5{\cdot}8\ x \sin(\theta + \delta)$ respectively.

Then $$h = \frac{O\,e}{F} = \frac{27{\cdot}2\,x\cos(\theta+\delta)}{7{\cdot}2\ x\cos(\theta+\delta)} = 3{\cdot}78.$$

Thus the central axis is 3·78 from the reference plane, and the resultants of the system are a force $7{\cdot}2\ x \cos(\theta + \delta)$ acting in the plane of the central axis or new reference plane, and a couple $5{\cdot}8\ x \sin(\theta + \delta)$.

The couple is always a minimum, and the central axis is, of course, always through the same point on the shaft, and therefore a reference plane which contains the central axis of the system is suitable for standard comparisons.

With this slight digression the consideration of the residual couples on the three-crank engine will now be proceeded with. The couple about an arbitrary plane must be taken first.

The moments of the residual vertical forces are—

$$M_1 \frac{v^2}{grs}\ a\ .\ \cos 2\,\theta \quad - \quad - \quad = x_1 \text{ say,}$$

$$M_2 \frac{v^2}{grs}\ b\ .\ \cos 2\,(\theta + 120^\circ) = y_1$$

$$M_3 \frac{v^2}{grs}\ c\ .\ \cos 2\,(\theta + 240\ ) = z_1.$$

Where *a*, *b*, *c* are the distances of the centres of engines from the reference plane respectively. Their signs must be according to the direction in which they are read.

Since the curves are all functions of $2\,\theta$ the same remarks apply to the sequence of the generating lines in the polygon as for the free forces. The resultant may be found and the curve drawn in a similar manner.

If it be desired to find the couple about the central axis plane the process explained above may then be proceeded with. The phase of the resultant couple will then be 90° in advance of the original one, and will, of course, have a different amplitude.

As there is no horizontal force there is, consequently, no horizontal couple.

If the engine be not completely balanced for functions of θ in the vertical direction, then, in addition to the residual resultants above, the resultant of the "single period" (θ) forces and couples must be found separately and their ordinates added to those of the "double period" (2θ) graphically, as explained in Case 2, par. 13.

15. The Four-crank Engine—*Preliminary.*—It will be evident from what has preceded that the three-crank engine (neglecting valve gear differences), where the three reciprocating lines are of equal weight, is in itself prefectly balanced without the addition of balance weights, so far as the free force is concerned, both for the primary and sec ndary periods. Further, that with three unequal lines of reciprocating weights it is possible to balance the primary vertical free force completely by a modification of the crank angles, so that the projected vectors on a reference plane form a closed triangle. The secondary period, however, might be c nsiderably out of balance, because it will be found impossible to close the secondary period triangle if the primary closes, and *vice versâ.* Fig. 309*a* shows a general case.

When the three lines are unequal, the valve gear can obviously be included without altering the nature of the results.

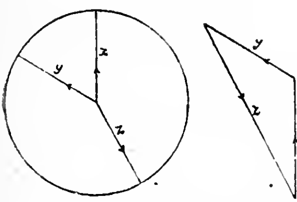


Fig. 309*a*.—Primary Forces.

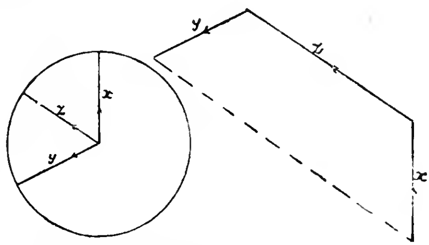


Fig. 309*b*.—Secondary Forces.

Without balance weights the unbalanced couple will necessarily be large, because the force triangles and couple triangles, either primary or secondary, cannot possibly close for the same crank angles. (*Euc.* 1-7.)

The four- (or more) crank engine offers much greater possibilities of balancing the reciprocating parts without balance weights, by virtue of the extra side to the polygons.

16. Four-crank Engine—*Yarrow-Schlick-Tweedy and Allied Systems.*—Given arbitrary positions of the cranks—for example, cranks forming a rectangular star—a four- or more crank engine may be balanced by means of balance weights and bob weights, as in the foregoing example of the three-crank engine.

It is, however, possible by arranging the cranks in a certain sequence and at angles differing a little from the rectangular star to obtain an inter-balance between the reciprocating parts themselves without the assistance of balance or bob weights.

The literature on balancing the four-crank engine is interesting, but exceedingly voluminous; most of it is of an academic nature, and capable of much abridgment.

The point so much overlooked is that (neglecting the complication of the valve gear, which as a matter of fact has in general practice been neglected) *it is only necessary to consider two lines as the bob weights to the other two.*

With this convention the problem is simplified, and can be worked out from the primary data without artificial aids.

It should be clear that the graphic method previously described is applicable, the example of the reciprocating parts of the three-crank engine being nothing more nor less than that of a five-crank engine if the bob lines be regarded as working lines.

There are, nevertheless, a few special points to consider when it is desired to balance the secondary periods so far as possible. A little reflection will lead to the following conclusions :—

1. The natural weights of the reciprocating parts combined with the natural cylinder spacing cannot necessarily be balanced for the couple (or more briefly for " tilting "), or even for the free force, because the magnitudes of the four sides of the polygon may not permit its closure.
2. The two inside reciprocating lines must be considerably heavier than the two outside lines, if the couple is to be balanced as well as the free force.
3. Given the cylinder spacing and minimum reciprocating weights (the two outside lines), it is possible by an arrangement of cranks to balance both the primary free forces and primary couple, but in general it is not possible to balance the secondary forces or couple completely. This will be evident from the fact that it is not possible to close two dissimilar polygons—the one the force, and the other the couple—with the angles of the one prescribing the angles of the other.
4. The main reciprocating weights should only participate in the solution of the crank angles. The rotating weights must be balanced by crank web balance weights in the usual way; and, further, as the valve gear design is not usually settled at the early stage when the crank-shaft must be ordered, and as the obliquity effect is very small, the valve gear may be included in the rotating system, the rotating balance weights being determined at a later stage in the design.

If it is permitted to re-adjust the cylinder spacings, it is possible to balance both primary and secondary forces together with the primary couple, but the four line weights must bear a definite relation to one another.

There are several methods of solving the problem, according to the data. Neglecting the obliquity of the connecting-rods, the simplest solution is that when the cylinder spacing is given together with one minimum line weight, which must be either of the outside lines.

The following is apparently due to Prof. Schubert, of Hamburg, and was introduced by Herr Otto Schlick in 1900,* the proof of the problem being given by the direct method instead of the simpler converse proof given below, which, however, is enough for the present purpose.

Set out the cylinder spacing to scale on a base line A D.

Take *any* point O above A D. Join O to A, B, C, D.

Draw C P parallel to O B and A P parallel to O D. Then O A P C is a mass quadrilateral giving obviously a balance of the free force if the four line weights be made proportional to the four sides.

* Paper on " Balancing of Steam Engines," *Inst. N. A.*, 1900.

If the four line weights be denoted by M_A, M_B, M_C, M_D, acting at A, B, C, D respectively, and if they be allocated as shown in the figure, the couple will be balanced as well as the force.

The crank angles will be as shown at point O, and care must be taken that the right and not the wrong weights are applied to the various cranks.

The diagram admits of simple proof.

Since the couple should be zero about any point, take moments about

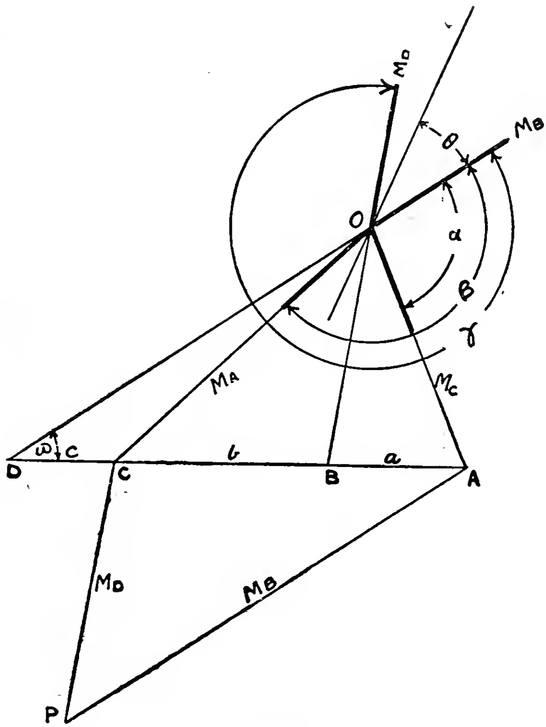


Fig. 310.

A; then, adopting the notation of the figure, we have for any value of θ measured from an arbitrary axis—

$$M_B a \cos\theta + M_0 (a+b) \cos(\theta+\alpha) + M_D (a+b+c) \cos(\theta+\gamma) = 0.$$

Put $\theta = 90°$, then—

$$M_0 (a+b) \sin\alpha + M_D (a+b+c) \sin\gamma = 0.$$

Now, $\dfrac{\sin\alpha}{\sin\omega} = \dfrac{a+b+c}{M_0}$, and $\dfrac{\sin\gamma}{\sin\omega} = -\dfrac{a+b}{M_D}$,

which establish the identity.

O is any point, and consequently there is an infinite variety of angles and corresponding weights that will give a balance for primary force and couple.

The selection of an appropriate point is a matter of judgment to suit the conditions, and, in general, it should be selected so as to give cranks as nearly at right angles as is possible consistent with the given minimum line weight.

The selection of the point O so that the secondary forces are balanced (that is, not neglecting the principal obliquity effect) is a problem that cannot be solved by direct mathematical analysis unless the cylinder spacing and reciprocating masses be symmetrical about the middle point.

The Yarrow-Schlick-Tweedy System is the name by which the symmetrical arrangement is known, and was introduced by Dr. Otto Schlick, of Hamburg, in 1900.

The following geometrical device will obtain point O to give a balance for primary forces and couple and secondary forces:—

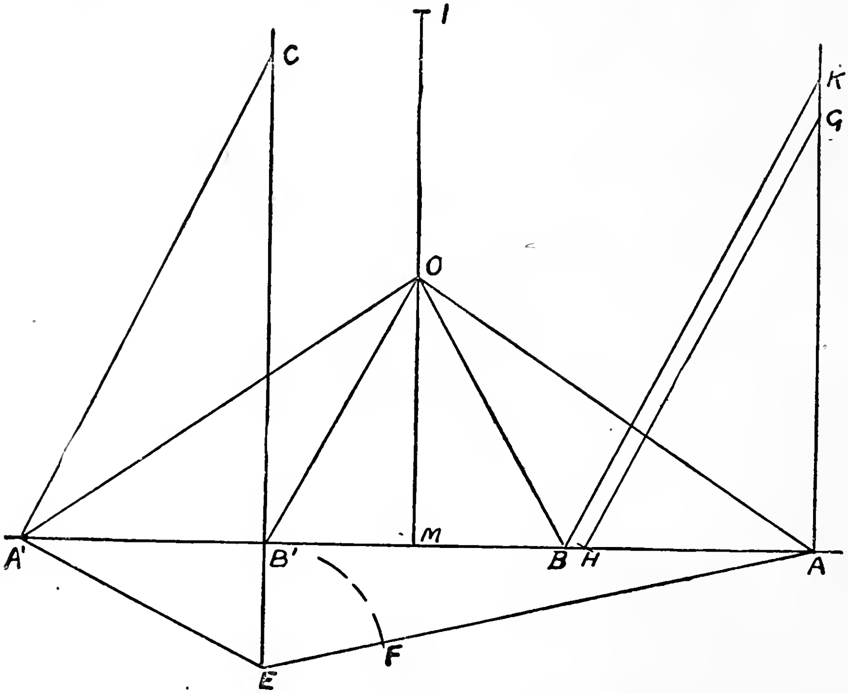


Fig. 311.

Erect perpendiculars at A and B′ (fig. 311).

Make A′ C = A′ B. Draw A′ E at right angles to A′ C. Join E A, and cut off E F = E B′.

Cut off A G = A F. With the compasses take B′ C as radius, and with centre G describe an arc cutting A A′ at H. Join G H = B′ C. From B draw B K parallel to H G. Make M I = B K.

Then M I bisected gives the point O the required apex of the characteristic triangles, as in fig. 310.

When the spacing is not symmetrical, and the two outside line weights are given, the secondary forces cannot necessarily be balanced, although the crank angles for the minimum resultant can be found by applying the ordinary method to two or three assumed conditions, and interpolating the best position.

The following example will illustrate this :—

Given cylinder spacing 42, 66, 37 (*v.* fig. 312*a*); outside line weights $M_A = 70$, $M_D = 80$.

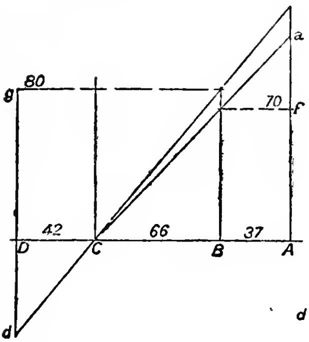


Fig. 312*a*.

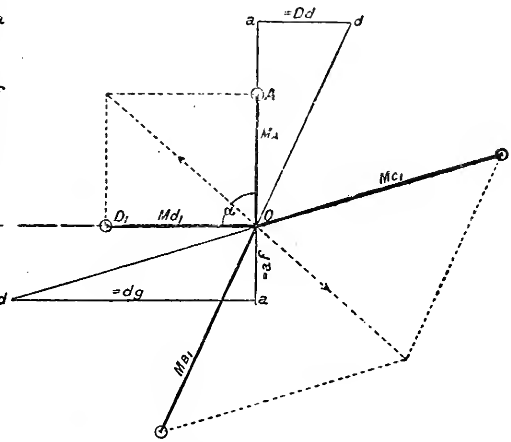


Fig. 312*b*.

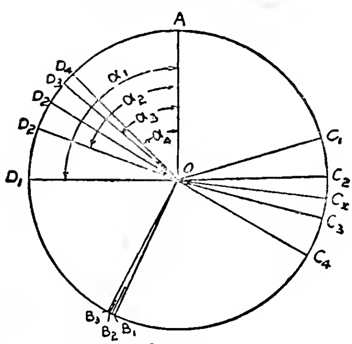


Fig. 312*c*.

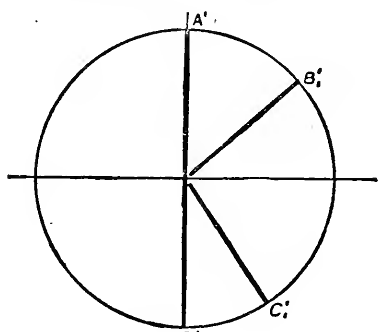


Fig. 312*d*.

Then, as in the previous examples, fig. 312*b* gives the magnitude and direction of M_{B_1} and M_{C_1} regarded as bob weights to M_A, M_{D_1} for an assumed crank angle A O D_1 or α_1.

The virtual crank angles for the secondary periods are as in fig. 312*d*, each real crank angle being, of course, doubled. O $a'\, b'_1\, c'_1\, d'_1$ (fig. 312*e*) is the polygon for the secondary forces, the forces being proportional to the line weights acting at the virtual crank angles.

This polygon does not close, and the secondary forces are, therefore, not balanced, the resultant being proportional to R_1.

Now, assume other values of the angle A O D, α_2, α_3, as shown in fig. 312c. From these are obtained the secondary resultant forces R, as in fig. 312e. In fig. 312e draw a curve through the extremities of the unclosed polygons.

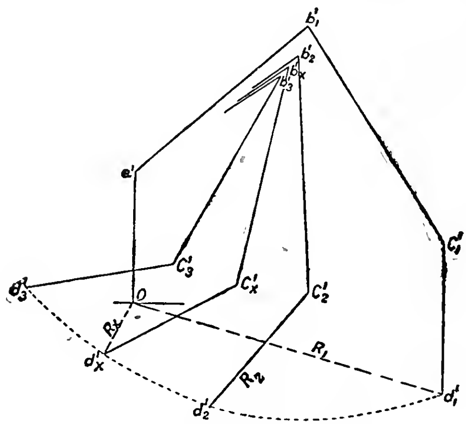


Fig. 312e.

Then R obviously has a minimum value at about point d'_x, from which it is easy to estimate the corresponding crank angles.

For this particular combination of weights and cylinder spacings it will be seen that it is not possible to balance the secondary forces completely;

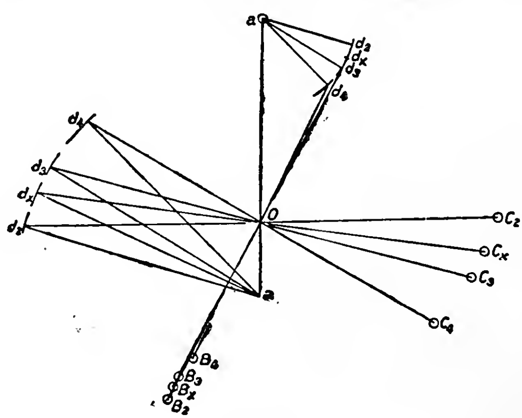


Fig. 312f.

but an increase of one of the lesser line weights or a slight modification to the cylinder spacing will effect it, and the balance can be obtained with very little trial and error.

The method outlined above, and followed also in the earlier part of the chapter, is one easily to be remembered, and is, therefore, available on any occasion.

When, however, these investigations are at hand the solution of any problem can be at once effected by means of diagram (fig. 313) prepared by Mr. C. E. Inglis, and given in his interesting paper at the Inst. Naval Architects, 1911.*

The meshed part of the diagram is a practical zone in which it is only possible to balance primary and secondary forces together with the primary couple.

The base line A D represents the cylinder spacing and the intersection of correspondingly numbered lines in the meshwork, the apex of the triangles of fig. 310—that is, point O.

For example, let the cylinder spacing be D-5, 5-14, 14-20; then the intersection of curves 5 and 14 give point O.

The directions of M_A, M_B, M_C, M_D are then as indicated by the dotted lines.

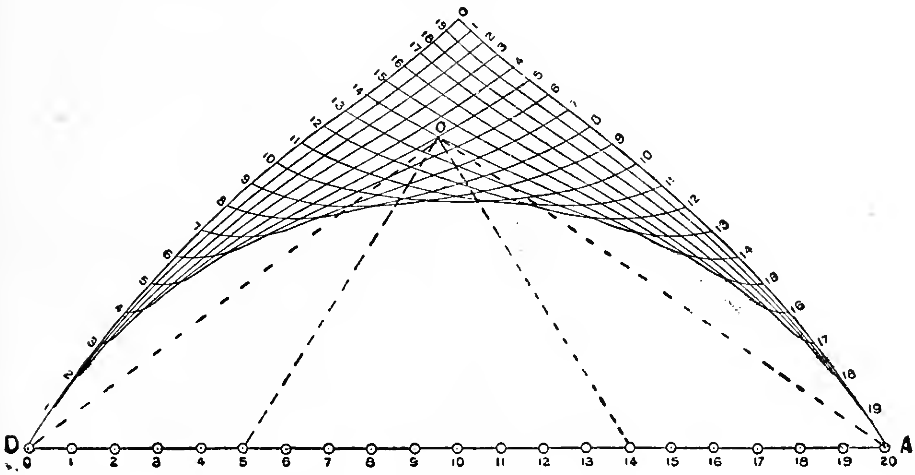


Fig. 313.

Further, the relative values of $M_{A \ldots D}$, determined by completing the polygon as in fig. 310, are the only possible ones which will give a balance for the given cylinder spacing and the three above static requirements.

Several simplifications will naturally occur in any problem if the cylinder spacing and line weights are symmetrical, as in the Y.S.T. system.

A very elegant solution was contributed by the late M'Farlane Gray to the Institution of Naval Architects (*vide Transactions*, 1900).

For more than four cranks the foregoing general methods will be found equally applicable, and in this respect it has the advantage of diagrams which are special to the four-crank engine.

Modified Principle of Balancing.—For an engine on any elastic foundation very large compared with the engine, and capable of vibrating in a manner

* Paper on "General Propositions and Diagrams relating to the Balancing of the Four-cylinder Marine Engine," *Inst. N. A.*, 1911.

analogous to a stretched string or beam spring, it is clear from physical laws that if the engine be placed at a node any unbalanced forces or couples should not be capable of exciting vibration in the foundations. Practically this holds quite true, and is only modified by the fact that the engine is of appreciable magnitude compared with that of the "loop" or distance between nodes.

If, however, the various line weights be selected so that the couples about the node are balanced, it is immaterial whether there is any free force left unbalanced, for it will cause no vibration.

This principle may be followed sometimes with advantage if the positions of the nodes of the ship are fairly accurately known, in which case it is only necessary to take moments about the nearest node, and close the polygon to determine the crank angles.

Modifications from a preliminary solution will probably be required to

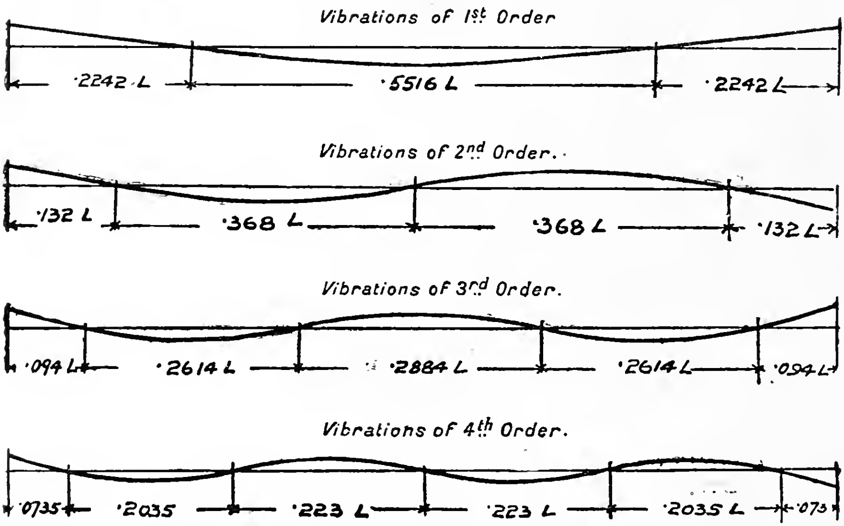


Fig. 314.

balance the secondary forces or couples. This method was followed some years ago by the author in balancing some new engines fitted in the twin s.s. "Cambridge," which ship certainly appeared to have less vibration than any similar twin-screw ships on the station fitted with three-crank triples. In this case the node was obtained by observations before the old engines were removed.

It is well known that sympathetic vibrations can be set up in an elastic body by any suitable agent, particularly if connected mechanically to it, vibrating with the natural periodicity of the body itself.

It therefore follows that the engine rate of revolution should never be allowed to correspond with the natural period of vibration of the ship herself.

The matter is, however, complicated by the fact that the significant vibrations of a ship may be of several orders, all the periodicities of which should be avoided. Thus, in the vertical plane a ship is capable of vibrating in an ascending series as indicated in fig. 314, those of the first two being usually the most important, on account of their generally greater amplitudes, although sometimes the higher orders have given some considerable trouble. For the first order Dr. Schlick finds that the position of the after node from the after perpendicular is ·231 to ·253 times the length of the ship for ships with very sharp lines, such as cruisers and despatch boats.

The distance of the fore node from the forward perpendicular varies from about ·31 to ·365 the length.*

For the second order there is very little data, and if the ship is not available at the time to supply it, the nearest estimate is that to be obtained by analogy with the known position of the various nodes in a prismatic rod.

These positions are indicated in fig. 314. The number of vibrations of the second order in the case of the rod is three times that of the first order, but in a ship it is found to be more nearly 2 to 1. The number of vibrations of the first order, according to Dr. Schlick,† are given very approximately by the following formula :—

$$N = \varphi\sqrt{\frac{T}{D\,L^3}},$$

where N = number of vibrations per minute.

T = moment of inertia in foot-tons of midship section about the horizontal neutral plane.

D = displacement of ship in tons.

L = length between perpendiculars in feet.

φ = a coefficient having values—

156,850 for vessels with very fine lines, such as torpedo-boat destroyers.

143,500 for large transatlantic passenger ships with fine lines.

127,900 cargo boats with full lines.

Clearly this method of balancing, in spite of its simplicity in the presence of reliable data, brings many undesirable complications; and an engine that is balanced within itself may seem a more satisfactory solution of the problem in general. At the same time, it must not be forgotten that large and long marine engines are far from being the rigid bodies they are assumed to be; and an assumption that the engine is an integral part of the ship, sharing its general elastic properties, is probably in most cases nearer the truth.

In any case, and whatever be the system of balancing, an examination into the possible coincidence of the proposed speed of revolution with at least the first and second orders of the natural vibration of the ship should be made.

17. The Effect of Inertia and Weight—*General Case, Supplement to the Remarks in the Introduction to this Chapter.*—It has been shown that the vertical inertia force at any instant is $-\frac{M v^2}{g r}\left(\cos\theta + \frac{\cos 2\theta}{s}\right)$.

* Schlick on "Vibrations of Higher Order, etc.," *Inst. N. A.*, 1895.

† Schlick, Paper on "Vibrations of Steamers," *Inst. N. A.*, 1894.

If this force is divided by the area of the piston the inertia pressure per square inch will be found. The effective pressure on the piston is, therefore, the algebraic sum of the steam and inertia pressures, and if the dead weight of the parts be included, the weight divided by the piston area must be added algebraically as well.

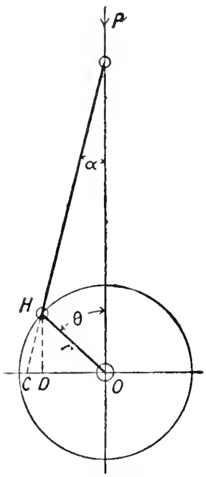


Fig. 315.

Considering the *inertia force* alone, M above is the total weight, including the rotating parts; but the revolving parts have also a horizontal acceleration, and the horizontal inertia force resolved tangentially to the crank path is equal and opposite to the vertical inertia force of the rotating parts resolved tangentially. This, of course, should be so, because the crank has no tangential acceleration. There is, therefore, no alteration produced in the effective twisting moment by the inertia of the rotating parts, and the weight of the reciprocating parts need only be used for obtaining the inertia pressure on the piston. M, in the above formula, will, therefore, be the weight of the reciprocating parts M_1.

It is easily shown that the twisting moment on the shaft due to an effective pressure on the piston P is—

$$\text{T M} = \frac{\text{P}\, r \sin(\theta + \alpha)}{\cos \alpha} \quad \text{(fig. 315)},$$

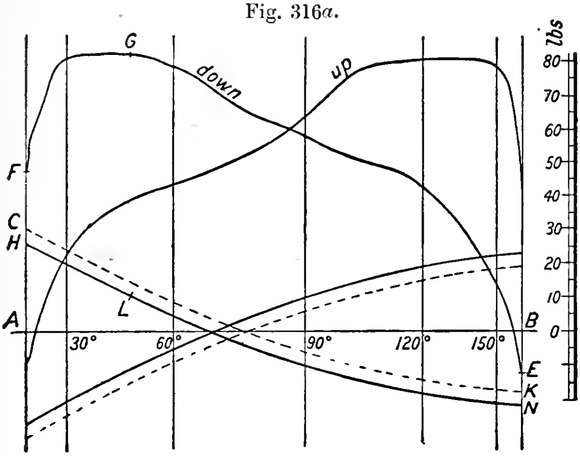
or, taking the second approximation to the series in terms of θ, as in the case of the acceleration (par. 4),

$$\text{T M} = \text{P}\, r \left(\sin \theta + \frac{\sin 2\,\theta}{s}\right).$$

In order to combine the inertia curve with the indicator diagram or steam pressure curve, the same spacing of ordinates for each must be adopted. If the inertia curve be drawn with indicator diagram spacing of ordinates (that is, the abscissæ are proportional to the distance of the piston from the end of the stroke) instead of abscissæ proportional to θ, it will assume the form shown in fig. 316*b* by the line C K. If $\cos 2\,\theta$ or the obliquity of the connecting-rod be neglected, the curve will be a straight line. The ordinates must, of course, be to the same scale as the steam pressure ordinates.

Considering the *dead weight* alone of the moving parts, the reciprocating parts having their motion controlled by the connecting-rod must necessarily have their weight added algebraically to the inertia force. But the rotating parts also cause a twisting moment on the shaft which, if M_2 equal their weight at radius r, is equal to $M_2 r \sin(\theta + \varphi)$, where φ is the angle between the resultant M_2 of all rotating masses and the crank. If there are no balance weights and valve gear be neglected, $\varphi = 0$. M_2 not being controlled by the length of the connecting-rod cannot be considered as being

Fig. 316a.



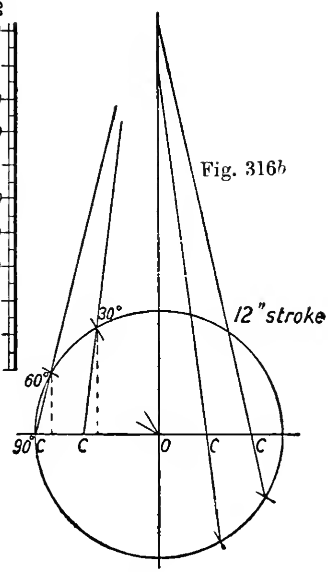
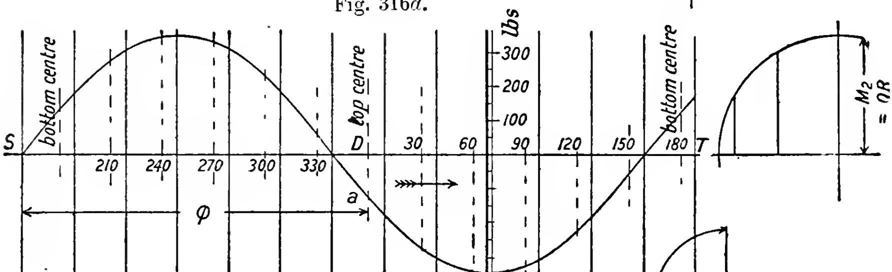


Fig. 316b

Fig. 316d.



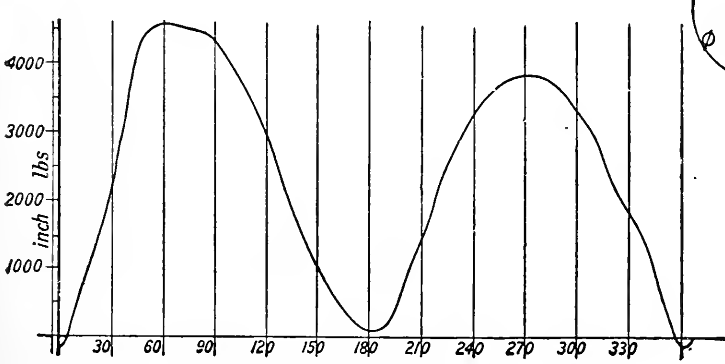


Fig. 316e.

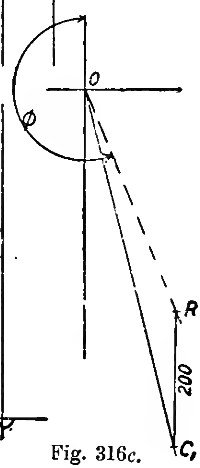


Fig. 316c.

distributed over the piston area (except for a connecting-rod of infinite length), and the effect of the weight of the rotating parts must, therefore, be considered apart from the indicator diagram.

Then (fig. 316*b*) the weight of the reciprocating parts being a constant quantity, set off $HC = \frac{M_1}{A}$ downwards for the downstroke and upwards for the upstroke, and draw a curve, H N, through H, parallel to the inertia line, C K.

Set off the indicator diagram from the straight back pressure line A B.

Then the effective pressure on the piston at the beginning of the stroke is not A F but H F, and at any other position of the piston the effective pressure, P, is given by the intercept between the curves F G E and H N. And similarly for the upstroke, due note being taken of the algebraic sense of the various pressures.

The total effective twisting moment at any instant is now—

$$TM = Pr\left(\frac{\sin(\theta \times \alpha)}{\cos \alpha}\right) + M_2 r \sin(\theta + \phi).$$

$Pr\frac{\sin(\theta + \alpha)}{\cos \alpha}$ may be calculated in a semi-graphical manner as follows:—
Let the crank be at O H (fig. 315), then

$$Pr\frac{\sin(\theta + \alpha)}{\cos \alpha} = P \times OC.$$

Also, when ϕ is 0, then

$$M_2 r \sin \theta = M_2 \times DO.$$

18. An example of Case 1 will now be considered.

Although it is very necessary to include the valve gear when balancing, it is rather an unnecessary elaboration to consider its influence on the twisting moment diagram, as it is so very small, unless, of course, the gear be abnormally heavy:—

Let M_1 = 400 lbs. (reciprocating).
M_2 = 200 lbs. (rotating).
Revolutions = 200 per minute.
12-inch cylinder, 12-inch stroke.
Connecting-rod 2 feet centres.
Position of fittings on shaft as in fig. 294.

Then

$$v = 10{\cdot}47 \text{ feet per second.}$$

$$v^2 = 109{\cdot}6.$$

$$s = \frac{24 \text{ inches}}{6 \text{ inches}} = 4.$$

$$\frac{M_1 v^2}{A g r} = 24{\cdot}1.$$

The values of the inertia pressure, etc., for every 30 degrees of the crank are tabulated below (fig. 316, *a*, *b*, *c*, *d*, *e*) :—

	$-\frac{M_1 v^2}{A g r} \times \cos\theta + \frac{\cos 2\theta}{s}$.	P = G L.	O C (Fig. 316*b*).	P . O C × A = Q_1.	$M_2 \times \sin(\theta + \phi)$.	$M_2 \sin(\theta + \phi) \times r = Q_2$.	T M = $Q_1 + Q_2$.
	Lbs.	Lbs.	Inches.	Inch-lbs.	Lbs.	Inch-lbs.	Inch-lbs.
0°	− 30·1	...	...	...	− 130	− 780	− 780
30°	− 23·9	60	3·64	24,700	− 270	− 1,620	+ 23,180
60°	− 9·04	73	5·82	48,000	− 345	− 2,070	+ 45,930
90°	+ 6·02	67	6	45,500	− 330	− 1,980	+ 43,520
120°	+ 15·06	60	4·52	30,660	− 225	− 1,350	+ 29,310
150°	+ 17·85	36	2·35	9,580	− 60	− 360	+ 9,220
180°	+ 18·08	...	...	...	+ 130	+ 780	+ 780
210°	+ 17·85	56	2·35	14,860	+ 270	+ 1,620	+ 16,480
240°	+ 15·06	61	4·52	31,200	+ 345	+ 2,070	+ 33,270
270°	+ 6·02	54	6	36,700	+ 330	+ 1,980	+ 38,680
300°	− 9·04	48	5·82	31,600	+ 225	+ 1,350	+ 32,950
330°	− 23·9	43	3·64	17,700	+ 60	+ 360	+ 18,060
360°	− 30·1	...	...	...	− 130	− 780	− 780

To find ϕ, let O C_1 (figs. 294*a* and 316*c*) be the resultant of the balance weights and C_1R equal weight of rotating parts (200 lbs.) then O R is their resultant.

Draw the curve O R sin θ (fig. 316*d*).

Set off S D = ϕ, then D a = O R sin $(\theta + \phi)$, or $M_2 \sin(\theta + \phi)$.

Fig. 316*e* is the effective twisting moment diagram.

19. The area of the final twisting moment diagram is proportional to the work done, and although its form has been modified by the inclusion of the inertia and weight, the area remains the same as if the steam pressure alone were considered.

This also follows from the principle of conservation of energy.

If the engine be balanced completely horizontally, and not vertically as in Case 1, the inertia and weight pressures from the reciprocating parts are the same as above, but the moment of the weight of the rotating parts is zero at all positions of the crank.

If the engine be balanced by bob weights (two) then the inertia of these must be considered exactly as if there were two extra cylinders in the engine, the only difference being that there is no steam diagram to combine with their inertia diagram.

The foregoing process applies equally well to a multi-crank engine.

The reciprocating weights and inertia pressures must be found separately for each cylinder, and the resultant horizontal weight-twisting moment for the whole engine combined with the resultant twisting moment due to the effective piston pressures.

20. Stresses due to Inertia.—As the stress in a given piece of machinery is due to the effective load applied to it, it therefore follows that inertia and dead-weight loads should be added algebraically to the steam loads when finding the stresses in those parts that are subject to them.

In very slow-running engines the effect of inertia is inappreciable, and may, therefore, be neglected, but in quick revolution and high-piston-speed engines its effect must not be lost sight of.

Starting at the piston, the first section to be dealt with is through the piston-rod thread. The inertia force is that due to the weight and acceleration of the piston, and not of the whole of the moving parts. The stress in the thread only comes into play on the upstroke, and the effective pressure on the piston is the intercept between the steam and weight lines as shown above.

Similarly, the stress at any section between the top of the piston and the centre of gravity of the connecting-rod may be found, the inertia line being constructed from the weight of the parts above the section.

As a rule, the factor of safety allowed for marine engines of the ordinary type amply covers the additional stresses due to inertia, and an engine that is well designed for static stresses may generally be considered to be strong enough to provide for dynamical stresses as well.

Referring to fig. 316*b*, it will be observed that the inertia line tends to make the effective pressure constant throughout the stroke.

For engines of normal scantlings and speed the maximum effective pressure usually occurs at about two-thirds the stroke from the beginning.

As the weights of moving parts are rarely available when an engine is being designed, and as there are also no indicator diagrams to refer to, the following approximate formula for finding the maximum effective pressure is given. This formula includes the total weight of reciprocating parts.

Let p_m = mean pressure of steam for that cylinder from the proposed I.H.P.,

p_f = maximum effective pressure,

L = length of stroke in feet,

N = revolutions per minute,

then $p_f = p_m + \cdot 0000386 \, N^2 L^2$.

The formula applies to the low-pressure cylinder for engines where there is only one low-pressure cylinder. If there are two low-pressure cylinders, then the medium pressure should be taken. It is not suitable for high-pressure cylinders of compound expansion engines, but this does not detract from the value of the formula, because the maximum loads generally occur in the above-meantioned cylinders.

Example.—To find the maximum effective pressure that may be expected on the low-pressure piston of a triple-expansion engine of 1,500 I.H.P., 150 revolutions, 54-inch low-pressure cylinder, 2 feet stroke—

$$p_m = \frac{1{,}500 \times 33{,}000}{3 \times 2{,}290 \times 2 \times 2 \times 150}$$
$$= 12 \cdot 00 \text{ lbs. per square inch.}$$

$$p_f = 12 + \cdot 0000386 \times 22{,}500 \times 4$$
$$= 12 + 3 \cdot 47$$
$$= 15 \cdot 47 \text{ lbs. per square inch.}$$

Example.—To find the diameter of the connecting-rod bolts (two) for the above engine, allowing a stress of 5,000 lbs. per square inch—

Maximum working load on piston = 2,290 × 15·47 = 35,426 lbs.

$$\frac{35{,}426}{2 \times 5{,}000} = 3 \cdot 54 \text{ square inches at bottom of thread}$$
$$= 2\tfrac{1}{2}\text{-inch bolts.}$$

21. The Design of Balance Weights.—As a rule, and more particularly with two- and four-crank engines, the great difficulty to contend with is being able to get the required mass in the space available.

With a uniform thickness and external radius there is a limiting length of weight (circular arc), beyond which any increase of mass makes the total mass less effective. The limit varies according to the shape of the weight and the angle which it makes with the crank.

When the weight is homogeneous and fairly opposite the crank its length should not subtend a greater angle than 120°, and in no case more than 140°, at the centre of the shaft. When inclined at a considerable angle to the crank, as shown in figs. 317 and 318, its length should not subtend more than 90°, on account of the long arm for attaching to the crank web—for any extension of length beyond these limits causes the centre of gravity to approach the centre of the shaft at a greater rate than the mass is increased.

For composite weights, as in figs. 317 and 318, the angles subtended with the centre of shaft may be increased about 5° above those given above.

The material of which balance weights are made depends, to a great extent, on the manner in which they are attached to the crank-webs.

In small engines, and when the weights of the moving parts are obtainable at the time the crank shaft is designed, and when the crank-shaft is a built one, the weights are very often forged as part of the webs.

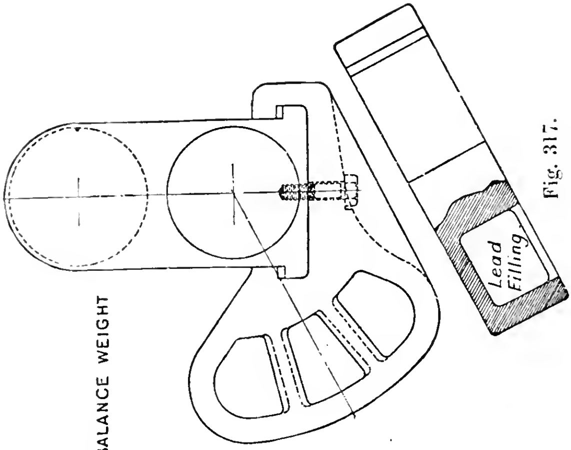
Circumstances, however, generally necessitate loose balance weights being fitted.

A neat and compact method of attachment is shown in figs. 317 and 318, and consists of forging small lips on the crank-webs, over which the weights are keyed.

The material the weights are made of, when this method of attachment is adopted, must necessarily not be of a short nature, and should either be forged or cast steel of a low tensile strength.

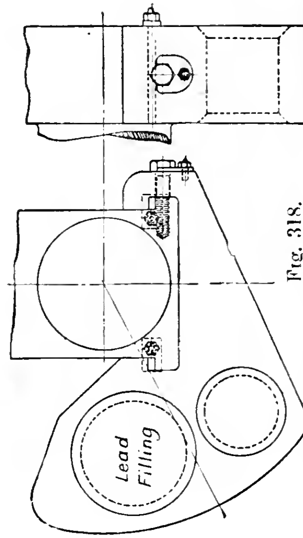
The Admiralty and a few other owners will only accept forged material, but there is no reason why a suitable cast steel should not be used for them. It has been used successfully by the author for many years. The advantage of a cast weight lies in being able to make it in the form of a shell which may be filled with lead, the materials thus being most advantageously disposed and the weight being of the smallest possible dimensions. It is also, generally, less expensive. This is shown in fig. 317. A forged balance-weight is shown in fig. 318. The most convenient method of manufacture is to shape them with a band-saw out of rolled slabs. Parts of the forging may be drilled out and filled with lead as shown, but at best this is an expensive addition and hardly worth the extra labour, as the quantity of lead that can be introduced is necessarily only a small fraction of that in fig. 317. Cast-iron weights are used to a great extent on high-speed land engines, and to a less extent for marine work. A cast-iron weight cannot be relied on if held on by lips as in figs. 317 and 318, and the usual method of attachment is by strapping it on as shown in fig. 319. The casting may be cored out and filled with lead if the small space available for it makes it advisable.

Cast-iron weights with its straps and fittings are, on the whole, much cheaper than either cast or forged steel, and there does not seem to be much reason why, beyond being somewhat clumsy in appearance, they should not be used to a greater extent than they are.



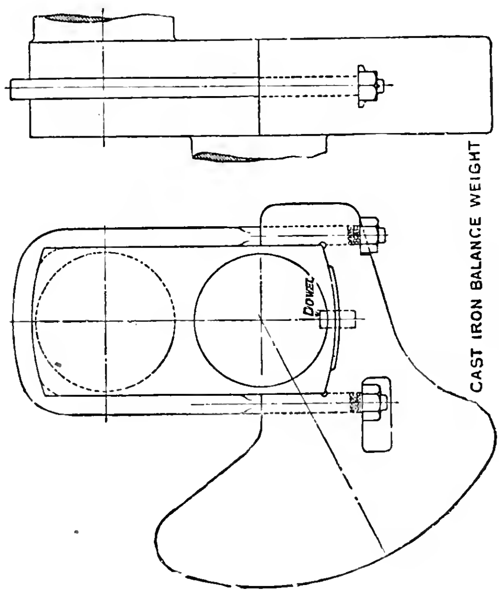
CAST STEEL BALANCE WEIGHT

Fig. 317.



FORGED STEEL BALANCE WEIGHT

Fig. 318.



CAST IRON BALANCE WEIGHT

Fig. 319.

In all cases the lips or straps must be made strong enough to take the centrifugal force of the balanceweight, together with the deadweight itself. If strong enough for this they will, in general, be found strong enough to resist the impulsive force on them, due to the engine being suddenly started or stopped.

It is not expedient in a manual of this kind to devote more space than the foregoing to the problem of balancing. Literature on this subject is exceedingly voluminous and fairly accessible. The aim of this chapter has been to place the elementary and practical principles before the student in such a manner that, should he desire it, he would have no difficulty in pursuing the subject more profoundly from the many highly mathematical papers extant.

The methods, short cuts, etc., of attacking the problems of actual practice will suggest themselves, as he proceeds, to the designer who has got a thorough grasp of the principles involved.

CHAPTER XXIX.

MATERIALS USED BY THE MARINE ENGINEER.

In every ship the weight of the machinery is of some consideration, and in many ships it is of the utmost importance that it be as light as possible. Reduction in weight can only be made by using material of superior strength, and by so designing the engine as to be able to employ such materials. The most marked advances in marine engineering, both in the size and quality of the work turned out, are in great measure due to the superior materials now available, and to the means possessed for converting them to the engineer's requirements. Steel, which was, only a few years ago, a comparatively expensive material, has now practically displaced iron, and that is largely on the score of cheapness. Not only is the price of steel now so low that it can compete successfully with iron, but steel can be produced in such large masses at such comparatively small cost, that heavy shafts* of this material made possible the immense engines recently constructed for the large ocean steamers and warships. Now steel alloys are supplied of such excellent quality, and with such high elastic limits, that for high-class engineering they are likely to displace ordinary mild steel, notwithstanding their prices being so much higher. Bronzes, too, have commanded attention, for not only do they hold their own for special purposes, but since certain kinds can be manufactured with as high a tensile strength as steel, and stand working hot or cold, they have displaced steel and iron for such purposes as propeller blades, parts of the engines, etc., in spite of the comparatively high price.

Cast Iron.—Although steel has superseded iron in many ways, cast iron still remains as the material most largely used by marine engineers. Its appearance is so well known as to need no description. The qualities of cast iron are usually connected with the district from which the ore is obtained. There also still remains a distinction from mode of manufacture, as "cold blast" is distinguished from "hot blast" in other ways than that of cost; but not much cold blast is made now.

All the pig iron of each district is, as a rule, divided into four to seven sorts, each of which is known by a number; that containing most free carbon, or *graphite*, is designated No. 1, that containing least Nos. 4 or 7.

No. 1 pig, when broken, exhibits a very coarsely granular fracture, having dark grey scales of considerable size; when melted "it runs very thin," that is, becomes extremely fluid, and on that account is used for ornamental castings and other work which requires a sharp outline, or is of very little thickness. It is not used much by marine engineers, except to mix with other kinds, or where extreme fluidity is necessary. Castings made from it

* Screw shafts are now made nearly 30 inches diameter and 60 feet long, weighing 64 tons in the rough state.

are very soft unless they are so thin that the metal has changed its character in the mould by chilling. The carbon here exists chiefly in the form of graphite scales, mechanically mixed as it were with the iron crystals.

No. 2 *pig* is not so soft nor so fluid when melted as No. 1, but is still not sufficiently close grained for general use.

No. 3 *pig* is that usually employed by the marine engineer for general purposes, as by adding some of No. 1, a mixture suitable for a complicated casting is obtained, and by adding to it some of No. 4, a harder and closer-grained casting can be made.

No. 4 *pig* is not much used for foundry purposes, except as a means of closing the grain and hardening the metal. It also differs from the other numbers in the appearance of the fracture; they all present a highly crystalline fracture of a distinct grey colour, and on that account called *grey iron;* it shows as a grey iron at the fracture, but the grain is much finer, and there is an absence of the coarse graphitic scales so strongly marked in the other three numbers.

Nos. 5 *and* 6 are not used at all for foundry purposes, but made for manufacture into wrought iron; on this account they are called *forge irons,* and as the fracture is still somewhat grey in places it is called "grey forge" to distinguish it from *No.* 7, which is called "white forge," as the fracture displays a distinctly crystalline structure, very hard, and silvery white in appearance. The carbon in this case is wholly combined with the iron chemically. Nos. 5 and 6 often present a mottled appearance, as if a grey iron and a white iron had been melted and imperfectly mixed; it is on this account that it is often known as "mottled pig."

It is often customary to generalise and recognise only three varieties of pig iron, viz.:—No. 1 as *grey* iron, No. 2 as *mottled* iron, and No. 3 as *white* iron; Nos. 2 and 3 being, of course, looked on as "forge" iron, and No. 1 as "foundry" iron.

In Great Britain there are many kinds of pig iron used by moulders, each called by the district where the ore is raised and smelted.

Scotch Iron is one of the best for foundry purposes; it is very uniform in quality, of good strength, and will mix well. There are various brands, those of Gartsherrie, Glengarnock, Eglinton, Carron, etc., etc., being best known. The No. 3 pig is most generally used by marine engineers, as it will run sufficiently fluid to make any casting, and judiciously mixed with scrap, it can be depended on for both closeness of grain and strength.

Cleveland Iron is used very much in the Cleveland district for general work; it is harder than Scotch iron, but does not possess so much strength, nor is it nearly so tough. A mixture of Nos. 1 and 3 is used for large work requiring strength, and this, when melted, possesses fluidity enough for the ordinary marine castings. Cleveland iron alone is not fit for large marine cylinders, columns, etc., but with the addition of hæmatite and scrap, to a mixture of Nos. 1 and 3, a good casting is obtained for general purposes.

Lincolnshire Iron is about equal in quality and general description to the Cleveland.

Staffordshire, Yorkshire, Derbyshire, and **Welsh Irons** are very good, and used in their own districts generally.

Shropshire Irons are very good, and from this county, Monmouth and Stafford the famous cold blast irons were obtained.

Cumberland Iron is made from *hæmatite* ore, and the pig generally goes by the name of "West Coast hæmatite." It is generally used for steel-making, but is also employed to improve other irons for foundry use. It possesses great strength and toughness, but cannot be used by itself for foundry purposes, as it does not run well when melted; 20 per cent. blended with good Scotch and English irons makes a strong mixture.

Cold Blast Iron.—Iron manufactured with air at the ordinary temperature is called by this name to distinguish it from the generality of irons which are made by heating the blast to about 700° F. The best known and most generally used cold blast irons were "Blænavon" and "Lilleshall," which owe part of their excellency, no doubt, to the good ore from which they are made. This iron possesses great strength with closeness of grain, and was used to close the grain and strengthen other irons, but is not now purchasable.

Iron Mixtures.—All cast iron is improved by re-melting, and the improvement continues until it has been re-melted as many as twelve times; after this it falls off in strength. No important casting should, therefore, be made from new iron only, and such as cylinders, pistons, cylinder covers, etc., should be made from iron wholly re-melted if the utmost strength is to be obtained. This rule, however, is only carried out when the thickness of metal is cut down to the lowest limit to save weight.

Cylinders, etc., which are subject to shock as well as to changes of temperature and severe strains, should be made of strong tough metal, and, since a good surface is required to withstand the wear of pistons and valves, the metal must have a close grain. Such cylinders, etc., used to be made of a mixture of one-third picked scrap, one-third best Scotch No. 3 pig, and one-third Blænavon or Lilleshall. If the casting is not large and complicated, the grain may be closed by adding picked scrap of a hard nature and increasing the amount of Blænavon. Some moulders add *hæmatite* to this mixture to increase the strength, but this is not often done, as it decreases the fluidity.

If the cylinder is to have liners and false faces for the valves to work on, it needs only to be *strong*; all *hard* materials, therefore, may be omitted.

Cylinder liners and false faces require to have a fair amount of strength, and be as hard as consistent with capability of being machined. Scrap iron of close grain and hard nature is selected to add to best Scotch No. 3 pig and Blænavon; if hard scrap cannot be obtained, some No. 4 pig may take its place; if necessary, even a small portion of "white" iron may be added to give additional hardness. Hardness and strength with closeness of grain may be obtained by mixing scrap steel (shearings and punchings from boiler-plates) to the extent of even 10 per cent. As the metal becomes much stiffer on the addition of the steel, only thick plain castings can be made with it. Propeller blades and bosses may with advantage be made with a mixture containing steel in lieu of hæmatite; the quantity of steel which may be added for this purpose depends on the founders being able to melt it.

Foundations and other large masses which are necessarily heavier than absolutely needed for strength may be made of a poorer mixture than suffices for cylinders; but as they are liable to shock, the iron must be of such a description as to resist this.

Specific Gravity of Cast Iron varies from 6·886 to 7·289; the average may be taken at 7·20. The weight of a cubic foot is, therefore, 450 lbs., and a cubic inch 0·26 lb. A square foot of it 1 inch thick weighs 37·50 lbs.

Strength of Cast Iron.—The following is the result of some careful experiments made at Woolwich Arsenal some years ago:—

	Minimum.	Maximum.	Average.
Tensile strength per square inch,	4·85 tons.	14·05 tons.	7·36 tons.
Transverse ,, ,, ,,	1·37 ,,	4·47 ,,	..
Torsional ,, ,, ,,	1·74 ,,	3·44 ,,	..
Crushing ,, ,, ,,	22·54 ,,	58·42 ,,	50·00 ,,
Shearing ,, ,, ,,	..	..	12·00 ,,

Good cast iron made by mixing qualities suitable for marine work should have an ultimate tensile strength of 25,000 lbs. and resistance to crushing of 110,000 lbs.; a bar 1 inch square and 36 inches long should deflect ½ inch without breaking with a load of 800 lbs. at the middle.

Wrought Iron.—This material, which is very nearly pure iron, is made exclusively in this country by the "indirect process"—that is, manufactured from cast iron by the process called puddling. The old methods of obtaining malleable iron direct from the ore, hence called the "direct process," are practised only by untutored tribes, or in regions inaccessible to general trade. The Siemens method of manufacturing steel is to some extent a reversion to the "direct process," and the product was sometimes properly called *ingot* iron, to distinguish it from iron made from the "*bloom*" which is obtained by puddling.

Rolled Bar Iron.—The "blooms" from the puddling furnace, after being squeezed or hammered, are rolled into bars, which are known as "*puddled bars*"; this bar iron is not used by engineers, as its tensile strength is sometimes as low as 9 tons per square inch; it is cut into pieces, which are piled crossways into a "faggot" or "pile," reheated, and rolled again into bars, and now called "*merchant bar*" iron.

Merchant Bars are not used by engineers for any very important work, as the iron is still of low tenacity and not very uniform in structure or quality. It was used for making gratings, ladders, etc., for fire-bars, bearers, etc.

Best Bar is made by reheating "faggots" of merchant bar iron, or good wrought-iron scrap "cross piled," and rolling it again into bars. Its strength is now much improved, and its quality more uniform, and it may be used for general smithing purposes. Its tensile strength, if made of good material, is about 24 tons per square inch on the average; the "Best" bars of some makers will withstand as much as 26 tons per square inch.

Best Best Bar is made by again rolling bars from faggots of selected Best bars. It has now a very uniform silky fibre, will bend double cold, and has a tensile strength of 26 to 27 tons; good specimens should elongate 25 per cent. at fracture with a reduction of area of about 50 per cent. Some kinds of iron have even a higher tensile strength than this, especially when rolled into round bars. Bar iron by cold rolling is increased in strength, but the elongation is reduced. Such iron is, however, seldom or never used now by engineers, except for the screwed stays of boilers.

Rivet Iron, when used, should be soft and of very good quality to withstand the work put on it in the process of riveting; its tensile strength is, therefore, somewhat low, about 24 tons per square inch, and its resistance to shearing is even lower than this, being from 20 to 22 tons per square inch.

This iron is generally made by rolling "piles" of selected scrap iron into bars, and for best quality rivet iron, the bar is rolled from "piles" of ordinary rivet iron.

Rolled Scrap Iron.—A superior quality of bar iron is made by "piling" shearings from boiler plates and rolling in the usual way; the quality of this iron, however, depends very much on that of the plates from which the shearings came.

Weight of Bars.—The specific gravity of bar iron of good quality is on the average 7·62; the weight of a cubic foot is, therefore, 476 lbs., and that of a cubic inch 0·276 of a pound. Yorkshire bar iron is somewhat denser, its specific gravity being 7·76; so that a cubic foot of it weighs 485 lbs.

Yorkshire Iron.—The best kinds of boiler iron were made in South Yorkshire, in the neighbourhood of Bradford and Leeds, and known as "best Yorkshire iron." Krupp, and some other German manufacturers, made iron plates of a quality equal to this; Swedish and Russian plates are very similar to it, and in some respects of superior quality.

It was of very uniform quality, and, although not possessing a very high tensile strength in direction of the grain, it was superior to other irons in strength across the grain; it was very tough, and had great elasticity, so that it was easily flanged and bent, and stretched very considerably before breaking. For these reasons, it was most valuable for boiler-making, and notwithstanding its high price (on the average three times that of ordinary boiler plates) it was always used for those parts of a marine boiler exposed to flame. Bar iron of this quality is still made in considerable quantities for smithing purposes in railway work, as well as for the screwed stays of boilers. It is practically as strong as boiler steel, and *stands alternating stresses better than ordinary steel;* it does not, however, take so good a screw thread. It is also used for crane chains and other high-class work, and was much esteemed for colliery work.*

Some best Yorkshire plates had a tensile strength of as high as 24 to 25 tons with the grain, and 22 to 23 tons across it; the elongation being 13·5 and 8 per cent. respectively. From some carefully made experiments, Mr. Kirkcaldy found the average strength of Yorkshire iron to be 21·3 tons with and 20·1 across the grain, the elongation being 16·7 and 11·2 per cent.; by annealing the plates the strength was slightly *reduced,* but the elongation was raised to 18·4 and 12·8 per cent. The elastic strength was also found to be 12·2 tons in tension, and in compression 11·5 to 13·3 tons.

Its specific gravity is 7·76; the weight of a cubic foot is 485 lbs., and that of a cubic inch is 0·281 lb.

Staffordshire Iron.—This iron, although slightly inferior to best Yorkshire, is still of high quality, and was used for boiler shells, domes, etc., and for such parts of the furnaces and chambers as were not exposed to the direct action of flame. It has a high tensile strength with the grain, but is not so strong across the grain as is the Yorkshire iron. Sir William Fairbairn, in 1861, found that some Best Best Staffordshire plates had an ultimate strength of 26·7 tons with and 24·47 tons across the grain, the elongation being 6·7 and 4 per cent.; that common Staffordshire plates had an ultimate strength of 22·7 tons with and 23·5 across the grain, the elongation being 5 and 4·35 per cent.

For purposes of calculation, the ultimate strength of Staffordshire quality

* The Board of Trade allows a working stress of 9,000 lbs. per square inch on such bar iron, provided it has been tested to 21·5 tons with an elongation of 27 per cent. in 8 inches.

boiler plates may be taken at 51,500 lbs. with, and 43,000 lbs. across, the grain for plates under ¾ inch thick; and at 50,000 lbs. with, and 40,100 lbs. across, the grain for plates over that thickness.

The specific gravity is 7·68, the weight of a cubic foot being 480 lbs., and that of a cubic inch 0·277 lb.

Staffordshire Bar Iron is very largely used for chains and similar gear, as well as for smithing purposes, and is very highly esteemed for them.

Iron Forgings.—Iron forgings, when used, are made from scrap iron, and their strength depends very much on that of the iron from which the scrap was cut. Sometimes forgings were made from new iron, but, since the general use of steel, this is seldom or never done now. The method of manufacture is similar to that described for making rolled bars from scrap; the scrap is sorted, piled, brought to a welding heat, and hammered into slabs; the slabs are piled one on the other, and reheated to form the forging required. The best description of forging is made by rolling the slabs into bars, so as to give the metal *grain;* the bars are then cut into short lengths, piled, and hammered again into slabs, which are piled, etc., as before, to form the forging. This rolling into bars, in addition to giving the iron fibre, tended to give a more uniform structure to the forging, and a homogeneity which cannot be obtained by the simple piling process.

The specific gravity of large forgings was about 7·63, so that the weight of a cubic foot was 477 lbs., and that of a cubic inch is 0·276 lb.

Steel.—Steel is used in the form of bars, plates, and forgings, and is also very generally employed for castings where great strength is required.

All steel was originally made from the best qualities of wrought iron by the process of "cementation"; this consists of exposing pieces of nearly pure iron to a high temperature in the presence of carbon only in a closed vessel for a considerable time, during which some of the carbon is absorbed by the iron, and is thus converted into a rough kind of steel, called *blister steel.* These pieces are broken, and sorted according to the appearance of the fracture after which portions are placed in a closed crucible, melted, and cast into ingots; it is now called *cast steel.* If the blister steel is piled, reheated, and hammered, or rolled into bars, it is called *shear steel.* The cast-steel ingots are worked into bars, which still retain the name "cast steel," but this is better known as *tool* steel, as it is now used almost exclusively for cutting tools. Tool steel containing about 1 per cent. of carbon is, of course, very hard, and has a very high tensile strength, ranging from 50 to 65 tons per square inch, with an average elongation of a little over 5 per cent. only; some of the milder kinds, such as are used for drifts, etc., have a tensile strength varying from 44 to 60 tons per square inch, with an average elongation of 13 per cent. *Spring steel* is still milder, having a tensile strength of about 45 tons per square inch, with an elongation of 18 per cent. Tempering in oil increases the strength considerably; Mr. Kirkcaldy found that a certain steel bar, whose strength when "soft" was 54¼ tons, when heated and cooled in oil had a strength of 96 tons.

Bessemer Steel.—The modern methods of making steel known as the *direct processes* obviate the necessity of using the comparatively expensive wrought iron, and are, therefore, capable of producing a very much cheaper material. In the Bessemer process there are essentially two operations, the conversion of molten *cast* iron into practically *pure* iron, and, by the addition of a small and definite quantity of manganese and carbon, the converting of pure iron into steel. Cast iron practically free from phosphorus and sulphur is melted and poured into the converter; a strong blast of air is forced through the molten metal, so that the carbon it contains burns and is consumed, and the temperature of the mass thereby raised; when the whole of the carbon is consumed, a small quantity of *ferro-manganese* (spiegeleisen or natural ferro-manganese was used originally), an iron alloy containing a known proportion of carbon and manganese, is added; the metal now has that small amount of carbon which causes it to differ from wrought iron, and that amount of manganese which seems to be so essential in making good steel. The metal is now run into ingot moulds, and allowed to cool for further use, or is kept in a "soaking pit," reheated and hammered, or rolled into the forms required. By the Thomas-Gilchrist process quite good steel can be made in a Bessemer converter from iron containing phosphorus; the phosphorus is absorbed by the converter lining of a *gannister*, which is prepared from magnesian limestone; the product is known then as *basic* steel, while that made by the original Bessemer process is called *acid* steel.

Siemens-Martin Steel.—The steel in this process is made in the hearth of a reverberatory furnace by exposing for a considerable time molten hæmatite or pig iron, or mixtures of cast iron and high-class pure (oxide) ores of iron, to the intense violet heat obtained by producer gas until practically the whole of the carbon disappears. If pig containing phosphorus is used, the furnace must be lined with *gannister* made from magnesian limestone on the Thomas-Gilchrist plan, so that the phosphorus may be absorbed almost wholly by the lining during the roasting process; *ferro-manganese* is added as in the Bessemer process, and the liquid mixture run into ingot moulds. Steel made in the ordinary way from good pig free from phosphorus is said to be by the *acid process*, while that made from pig iron containing considerable quantities of phosphorus in a furnace or converter lined with magnesian limestone gannister is called *basic steel*, and said to be made by the Thomas-Gilchrist or *basic process*. Basic steel is generally softer than ordinary steel, but excellent metal is made by this process, having an ultimate tensile strength of not more than 25 tons, with a stretch of over 30 per cent.; it can be bent cold two double, and is soft to work and easily welded. Large quantities of this material are now made at Middlesbrough, North Lincolnshire, etc., where there is an unlimited supply of suitable iron, and used by shipbuilders, tank and bridge builders, etc. Basic steel of much higher tensile strength can be now obtained and used with satisfaction.

At the present time steel plates, bars, and forgings are made almost exclusively from ingots run from either Bessemer converters or Siemens furnaces, and the basic is the kind most in use by engineers.

The Steel for Boiler Construction,* however, is made by the Siemens-Martin process exclusively. The ingots are reheated and hammered into slabs, which are again reheated and rolled into bars or plates. It is necessary that the material of which a boiler is constructed shall have very considerable elasticity as well as strength, and since best Yorkshire iron stretched to as much as 18 per cent. before fracture no steel is used which is not equal to this.

* Boiler plates may be *basic* or *acid*, but in every case the steel must come from an open hearth furnace.

It is found that the lower the ultimate strength of good steel is, the higher is its elasticity, so that while plates of this material, having an ultimate elongation of 20 per cent., possess a tensile strength of 35 tons per square inch, those having an ultimate strength of only 26 tons will stretch 30 per cent.* The former is very suitable for the shell-plating of boilers, as its strength is nearly 50 per cent. higher, and its elasticity nearly double that of the iron formerly used by boilermakers; the latter does eminently for the internal parts of a boiler which require to be of a somewhat softer material, that it may be flanged, etc., with ease, and stand the rough usage of the boilersmiths with safety. For this purpose plates are used which have a tensile strength of 26 to 28 tons per square inch, with an ultimate elongation of about 25 per cent. For corrugated furnaces a milder steel still is used, for its limit of strength is 25 tons, and it stretches to 30 per cent. in 8 inches. Shell-plates are now made with a tensile strength of 35 to 40 tons, and a stretch of 20 to 25 per cent. with the grain, and 20 per cent. across the grain.

Subjoined are the tests required by the various authorities for boiler steel. It will be seen that no two of them agree, which is much to be regretted; it is in the interest of both science and economy that they should all agree :—

Admiralty Tests of Material.—All steel to be made by the acid open hearth process. Every plate, etc., used is to be tested, and must comply with the requirements stated below :—

TABLE XCII.—ADMIRALTY TENSILE TESTS.

Description of Material.	Minimum Ultimate Tensile Strength, Tons per square inch.	Maximum Ultimate Tensile Strength, Tons per square inch.	Minimum Elongation in 8 inches per cent.
			per cent.
Not exposed to flame and not flanged,	27	30	20
Exposed to flame and flanged,	24	27	25
Rivet bars,	24	27	25
Steam-pipe plates,	24	27	33
Corrugated or ribbed furnace,	23	25	27
Tube forgings (annealed),	21	24	27
Pieces cut from tubes (annealed),	...	26	27

For bending tests the specimens are heated to a low cherry red, and then cooled in water at 82° F. Strips of plate $1\frac{1}{2}$ inches wide must bend double in press, inner radius being $1\frac{1}{2}$ times thickness of plate. For pieces of rivet bar inner radius to equal radius of bar; and for strips from tubes, $\frac{1}{2}$ inch.

Board of Trade Tests of Material.—Strips 2 inches wide should be cut from at least one of every four ordinary plates, from each end of each plate over 15 feet in length, and from each corner of each plate over 20 feet × 6 feet, or over $2\frac{1}{2}$ tons in weight; where more than one test piece is taken from a plate the *mean* result is to be adopted.

* The ultimate tenacity in tons plus the percentage of stretch in 8 inches should not be less than 52 for boiler steel. For forgings, Lloyd's Register requires this to be 57.

TABLE XCIII.—Board of Trade Tensile Tests.

Description of Material.	Minimum Ultimate Tensile Strength, Tons per square inch.	Maximum Ultimate Tensile Strength, Tons per square inch.	Elongation in 10 inches per cent.
Plates not exposed to flame,	27	32	20 per cent. in 8 inches.
Plates that are exposed to flame,	26	30	Not less than 23 per cent. for annealed plates.
Rivet bars,	26	30	Not less than 25 per cent.
Stay bars,	27	32	Not less than 20 per cent. and in combustion chambers 18.
Tube strips,	26	30	About 25 per cent., not less than 20 per cent.
Rivets,	27	32	Contraction of area about 60 per cent.

N.B.—Shell plates up to 36 tons tensile may be used by arrangement.

For bending tests the specimens should be 2 inches wide and 10 inches long, and should be bent double, the inner radius being 1½ times the thickness of the plate; for plates not exposed to flame, the tests should be made on the plate in its ordinary condition, and for plates that are exposed to flame the specimens should be heated to a cherry red, and cooled out in water at 80° F. before bending.

Lloyd's Tensile Tests of Plates. etc.—Steel plates for shells and girders 28 to 32 tons per square inch; plates for flanging, welding, or exposure to flame 26 to 30 tons. Material for purposes where tensile strength is unimportant bend tests only are required. The elongation of test pieces must not be less than 20 per cent. in 8 inches for material 0·375 inch and over in thickness whose tensile strength is 28 to 32 tons: and not less than 23 per cent. when it is 26 to 30 tons and 0·375 inch and upwards in thickness.

Stays, angle and tee bars, 28 to 32 tons tensile, with 20 per cent. elongation, while screwed stays are 26 to 30 tons and 23 per cent. Rivet bars, 26 to 30, with 25 per cent.

Material under 0·375 inch thick, the extension may be 3 per cent. below the above.

Test pieces, not less than 1½ ins. wide, from plates or bars to stand bending to a curve, the inner radius not greater than 1½ times thickness, when in normal condition and after having been heated to a low cherry red and quenched in water at 80° F. Steel rivets to be bent cold and hammered until the shank is bent round on to itself without fracture on outside of bend, and heads flattened hot to 2½ times diameter of shank.

One tensile, and either one cold or one temper bend test to be made from every plate; but for plates exceeding 2½ tons in weight, one tensile test from each end, and a cold bend test from one end and a temper bend test from the other are required.

British Corporation Boiler Steel.—The quality of steel to be used in the construction of boilers must be as follows, viz.:—

Plates intended for the cylindrical shells and butt straps, and bars intended for stays, are to have an ultimate tensile strength of not less than 28 tons, or more than 32 tons per square inch, with at least 20 per cent. extension in 8 inches of prepared section. Plates which have to be flanged or welded, plates and stays for combustion-chambers, plates for furnaces, and bars intended for rivets are to be from 26 to 30 tons per square inch in tensile strength, with not less than 23 per cent. extension in 8 inches, but with rivet bars the extension is to be not less than 25 per cent. in 8 inches, and 31 per cent. in 3½ diameters.

The sample pieces for temper bend tests are to be heated uniformly to a dull red, then cooled in water the temperature of which does not exceed 80° F., and the pieces in both cold and temper bend tests must withstand, without fracture, being bent double to a curve, the inner radius of which does not exceed $1\frac{1}{2}$ times the thickness of the piece.

All plates which have been welded, locally heated, or in which rivet holes have been punched, and steel stay bars which have been worked in the fire must be subsequently annealed, and in no case are steel stays to be welded. Number of tests same as for Lloyd's, see above.

John Spencer & Sons, Newburn, and David Colville & Sons, Motherwell, manufacture plates and forgings from patent silicon steels which have an ultimate tensile strength of 40 tons, with an elongation of 30 per cent. in 2 inches and a limit of elasticity of 25 tons. Such plates, however, cost a little more to manufacture than the ordinary Siemens steel, but they repay the boilermaker for it.

Ordinary mild bar steel, such as used for the stays of boilers, for bolts, studs, etc., has a tensile strength of 26 to 32 tons, with an elongation of 25 to 20 per cent., and a harder steel for pins, etc., has 35 to 40 tons, with 15 per cent.

The specific gravity of mild steel plates and bars is about 7·86; the weight of a cubic foot is 491 lbs.; that of a cubic inch is 0·284 lb., and that of a square foot, 1 inch thick, 40·94 lbs.

Steel Boiler Plates can be obtained now of large size, as follows:—

Plates $\frac{1}{2}$	inch thick, 10	feet wide, or 40	feet long, but area not to exceed 240	sq. ft.
,, $\frac{9}{16}$	,, 10	,, 50	,, ,, ,, 260	,,
,, $\frac{5}{8}$	,, 11	,, 50	,, ,, ,, 300	,,
,, $\frac{3}{4}$	,, 12·5	,, 50	,, ,, ,, 325	,,
,, $\frac{7}{8}$	,, 12·5	,, 50	,, ,, ,, 350	,,
,, 1	,, 12·5	,, 50	,, ,, ,, 430	,,
,, $1\frac{1}{8}$	,, 12·5	,, 50	,, ,, ,, 450	,,
,, $1\frac{3}{16}$	,, 12·5	,, 50	,, ,, ,, 430	,,

Plates $1\frac{1}{4}$ may be up to 415 square feet, those $1\frac{3}{8}$ to 380, those $1\frac{1}{2}$ to 340, and those $1\frac{5}{8}$ to 300 (J. Spencer & Sons).

Plates $1\frac{3}{4}$	thick may be 12	feet wide or 30	feet long, and the area to 288	sq. ft.
,, 2	,, 12	,, 25	,, ,, 250	,,
,, $2\frac{1}{2}$	,, 12	,, 25	,, ,, 200	,,

Steel Flat Bars are rolled from $\frac{1}{2} \times \frac{1}{8}$ thick to 8 inches $\times$ 4 thick.

Square Bars from $\frac{1}{4}$ inch to 6 inches, and with round corners to 8 inches.

Round Bars from $\frac{1}{4}$ inch to 8 inches, and from 10 inches to 16·5 inches by Spencer's up to 30 feet long.

Thin Plates as low as $\frac{1}{4}$ inch can be obtained 6 feet wide or 30 feet long, but the area must not exceed 120 square feet.

Round Plates $\frac{1}{4}$ thick may be obtained 6·5 feet diameter; $\frac{1}{2}$-inch plates 11 feet diameter; $\frac{3}{4}$-inch plates 12 feet; $\frac{3}{4}$-inch and upwards 13 feet diameter.

The basis price for boiler plates is about £7 per ton,* tested to pass Board of Trade, Lloyd's, or other Registry, but that price does not include very large or very thick ones, and is for the ordinary commercial boiler steel of 28 to 32 tons tensile strength. If a minimum of 29 tons is required the extra price is 10s. per ton, if 30 tons it is 20s. Should a high tensile steel be necessary, such as 35 tons per square inch as a minimum, such plates will cost £6 extra to the basis.

If more than 15 per cent. of the plates are not truly rectangular there is an extra charge for all beyond this quantity of 25s. per ton.

* In normal times, in 1918 it is much higher.

Plates will be supplied up to $1\frac{1}{2}$ inch thick at basis price, provided they are not over 8 feet wide; over that width, there is an extra charge of 2s. 6d. per 3 inches additional width, so that the 12·5 feet wide plates will cost 45s. per ton beyond basis.

Plates which are over	$1\frac{1}{2}$	inches to	$1\frac{5}{8}$	inches thick will cost	10s.	per ton extra.
" "	$1\frac{5}{8}$	"	$1\frac{3}{4}$	" "	20s.	"
" "	$1\frac{3}{4}$	"	2	" "	30s.	"

So that a plate 2 inches thick will cost 60s. extra per ton. If very mild steel is required, so that the maximum limit of tensile strength is 27 tons, a charge of 10s. extra is made, and if as low as 25 tons maximum it is as much as 40s. It will be seen, then, that if very large thick plates are used in boiler construction, their cost will be somewhat high; but, on the other hand, the saving in labour and materials will more than compensate for it, besides permitting of a superior boiler from the users' point of view.

Steel Forgings.—Shafts, piston- and connecting-rods, valve-rods, gudgeons, etc., are now made of steel forged from ingots manufactured by the Siemens process. The steel is, of course, generally of a mild kind, and while possessing properties very similar to those of the rolled bars and plates, is not always quite so uniform in structure and strength.

There is little doubt that the ultimate strength of marine shafts, when made of steel, does not exceed 35 tons on the average, and those over 12 inches diameter cannot be depended on for a higher average than 30 tons, however well forged. The material near the centre of a steel forging of large size remains but partially affected by hammering, so that the larger the diameter of the shaft, the less will be the average strength of the material composing it. Forgings as made with a hydraulic press are freer from suspicion, but even they are not good near the axis when of large size.

Small steel forgings may have as high a tensile strength as bars rolled from similar ingots, but as a rule they have not so great elasticity, and it is safer, therefore, to suppose them to be 10 per cent. weaker, for purposes of calculations, although, as a matter of fact, their tensile is sometimes higher. Lloyd's require the tensile and percentage of elongation to be together 57.

Steel Castings.—Many parts of a marine engine which were formerly of forged iron, are now made with advantage of cast steel; other parts which, for convenience of manufacture, were made of brass, are often of this material. It has superseded cast iron in many parts which must of necessity be cast, and as the cost of production of cast steel is reduced, and the soundness of it improved, so will the demand for it increase.

The chief obstacle to further employment is usually this uncertainty as to the soundness of the castings. Continued use, however, is dissipating the prejudices which once existed against its application, and the demand for these castings has caused some manufacturers of them to give the closest attention to their production, the day should not be far distant when a steel casting will command the same confidence as to its soundness that now obtains for iron castings. Indeed, an iron casting may be really more treacherous than one of steel, because blowholes and spongy places are always near the surface of the latter, and can often be detected, while those in the iron castings are quite hidden. If a steel casting is machined, so that the faulty places are cut away, the part remaining may be depended on as quite sound.

Propeller bosses, foundations, columns, levers, crossheads for piston-rods, pistons of all sizes and shapes, link-motion blocks, eccentric straps, worms, wheels, etc., are now very generally made of cast steel; also large crank-shafts have been made of cast steel, and did their work very satisfactorily, even when so large as 15 inches diameter; connecting-rods, also, have been made of this material, and the economy of production of such parts when thus made in quantity is beyond all doubt.

The probable *average* strength of best steel castings is about 28 tons per square inch, with an elasticity of about 25 per cent. in 2 inches; greater strength is easily obtained with steel castings, but the elasticity will then be much lower. in fact, it is only by using very carefully selected materials, and by annealing after casting, that *so low* a strength can be obtained. The ordinary steel castings have an average ultimate strength of about 32 tons, with an elongation of about 18 per cent. when sound. To allow for unsoundness the ultimate strength may be assumed to be only 28 tons for the harder varieties, and 25 tons for the softer ones; for propeller blades it should not be assumed to be higher than 24 tons, although the sound parts of such castings are often found to have a strength of over 30 tons per square inch.

Nickel Steel.—This material, which is an alloy of steel and nickel, or, to be more precise, of iron, carbon, manganese, and nickel, is often used where lightness is of first importance in engineering structures, or when the strongest possible material is required. Great care, however, has to be taken when forging and otherwise treating this material in the hot state to avoid seriously injuring its physical properties. It is, therefore, better to have the forging, etc., delivered from the maker's works ready for machining, and in all cases it should be most carefully annealed. It is also advisable to have it oil-tempered as well as annealed to get the full advantages from this splendid alloy. Sir Wm. Beardmore read a very interesting paper to the Institution of Naval Architects, in 1897, on the nickel-steel alloys his firm was making for boiler plates; with an ultimate strength of 74,000 lbs., and an extension of 23 per cent., the elastic limit was 43,000 lbs.; and a stronger quality whose ultimate strength was 114,000 lbs., with 25 per cent. extension, the elastic limit was 64,000 lbs. Forgings on a large scale have been made in this country by Messrs. Vickers, Sons & Maxim, Beardmore, and others for marine work as well as armour plates; while in America, at the Bethlehem Iron Works, this material is very largely used for marine work. Steel having 0·25 to 0·45 of carbon seems best for this alloy, and for marine purposes 3 to 5 per cent. of nickel is used with it. Less than 3 per cent. nickel is of no advantage.*

Nickel steel as made in Sheffield and in Germany is generally as follows:—

Place of Manufacture.	Carbon.	Silicon.	Manganese.	Nickel.
Sheffield, ...	0·280	0·123	0·516	2·95
Germany, ...	0·310	0·112	0·625	4·18

Forgings having 0·29 per cent. of carbon and 3·25 of nickel have an ultimate strength of 85,000 lbs., with 30 per cent. elongation, and an elastic

* Chrome nickel steel is now largely used for light forgings where great strength is necessary.

limit of 54,000 lbs.; with 3·75 per cent. of nickel and oil tempered, 90,000 lbs., with 22 per cent. extension, and 60,000 lbs. elastic limit; with 4·5 to 5 per cent. the elastic limit is as high as 80,000 lbs., and the extension 20 per cent. after careful oil tempering and annealing. The coefficient of expansion of an alloy containing 25 to 30 per cent. of nickel is exceedingly low, and its electrical resistance is 40 times that of copper; it is also nearly incorrodible.* Adding 3 to 3½ per cent. of nickel increases the cost of a steel by about 7s. per cwt. This material stands percussion shocks exceedingly well, and also resists vibratory stresses equally well. Iron bars stressed to 40,000 lbs. stood only 59,000 alternations, mild steel 170,000, high carbon (0·35 per cent.) 317,000, while nickel steel (0·25 C, 3·25 Ni) stood 1,850,000, and 5½ per cent. nickel as many as 4,370,000 alternations.

Chrome Vanadium Steel is now extensively used for engineering purposes on shore and to some extent on board ship, where its high qualities make it a most useful addition to the means for effecting special ends where price is of secondary importance. The addition of quite small quantities of these metals, either singly or together, to the ordinary carbon manganese steels of commerce improves them immensely, as may be seen by examining the following table:—

	Elastic Limit.	Ultimate Tensile Stress.	Elongation in 2 inches.	Reduction of Area.
	Tons per square inch.	Tons per square inch.	Per cent.	Per cent.
Carbon manganese steel, crucible, . .	16·0	27·0	35·0	60·0
„ „ + 0·5% chromium,	22·9	34·0	33·0	60·6
„ „ + 1·0% „	25·0	38·2	30·0	57·3
„ „ + 0·10% vanadium,	28·5	34·8	31·0	60·0
„ „ + 0·15% „	30·4	36·5	26·0	59·0
„ „ + 0·25% „	34·1	39·3	24·0	59·0
„ „ + 1·0% chromium + 0·15% vanadium,	36·2	48·6	24·0	56·6
„ „ + 1·0% chromium + 0·25% vanadium,	49·4	60·4	18·5	46·3
Open-hearth „ plain, . . .	17·7	32·2	34·0	52·6
„ „ + 1·0% chromium + 0·15% vanadium,	34·42	52·6	25·0	55·5

Under torsion some bars of vanadium chrome steel ¾ inch diameter and 6 inches long had an ultimate shearing resistance of 40·6 tons, with an elastic limit of 17·4 tons, and the final angle through which they were twisted was 1,623 degrees or 4·51 turns.

Some samples of this material containing the following, viz.:—

Carbon, ...	0·297	per cent.
Silicon, ...	0·059	„
Manganese, ...	0·394	„
Chromium, ...	1·066	„
Vanadium, ...	0·169	„

were subject to a series of most exhaustive tests by Captain Sankey and

* An incorrodible chrome steel is now made having an ultimate tensile strength of about 60 tons, with a yield point of about 50 tons and elongation 18 to 20 per cent.

Mr. Kent Smith, of which the following is an interesting extract, as showing how it is affected by heat treatment:—

Condition.	Yield Point.	Ultimate Stress.	Ratio Yield to Net.	Elongation in 2 inches.	Reduction of Area.	Number of Alternations.
The steel as delivered from the rolls, . .	36·9	54·1	0·68	24·0	44·9	1,906
After annealing for half-hour at 932° F., . .	36·9	53·8	0·68	24·7	48·9	...
After being water-quenched from 932° F., .	35·6	53·5	0·67	25·6	50·7	...
After being oil-quenched from 932° F., . -	35·8	54·1	0·67	26·0	49·6	...
After being water-quenched from 1472° F., .	60·3	74·6	0·81	7·5	16·6	174
After being oil-quenched from 1472° F., . .	37·0	54·5	0·68	22·0	35·2	296
After annealing for half-hour at 1472° F., -	21·1	39·0	0·54	34·5	53·1	2,237
After being oil-quenched at 1598° F., and reheated to 662° F., .	49·8	59·3	0·85	23·0	50·8	1,314
After being water-quenched at 1652° F., .	56·6	78·0	0·72	10·0	21·5	...
After being oil-quenched at 1652° F., . .	47·9	60·9	0·79	18·5	39·1	...
After being oil-quenched at 1652° F., and reheated to 1112° F., .	43·4	53·6	0·81	22	56·6	1,938

Soft Steel for Solid-drawn Tubes is usually of Swedish make or made elsewhere of Swedish ores, which are remarkably pure and free from phosphorus and sulphur. Such steel usually contains about 0·127 per cent. of carbon, 0·016 of silicon and 0·332 of manganese. It is always ductile and soft, so that it can be drawn down cold to almost any degree of thinness required, while being free from any blemish, or even mark, that would raise a reasonable doubt as to its soundness.

Manganese Steel.—The invention of Sir Robert Hadfield is of interest, although it is not so generally useful to the engineer as is to be desired. That best known is an alloy containing 11 to 14 per cent. of manganese, and is largely used for the wearing parts of dredgers, crushing machinery, and ash ejectors. It is very tough and very hard, so that while it can be forged easily and bent readily when cold, it cannot be machined. In the forged condition it has a tenacity of 60 tons per square inch, with an elongation of 35 to 40 per cent. in 8 inches. At present articles are forged to shape or ground, but it is hoped that soon some method of machining this valuable material will be found.

Copper.—This metal is used to form alloys more frequently than employed by itself; it is too soft for general purposes; but as it is so much improved by the addition of even small quantities of other metals, it is perhaps, next to iron, the most important of the metals used by the marine engineer.

In commerce there are three distinct qualities of ingot copper dealt with, viz. :—(1) G.M.B. (good merchantable brand), a general term used as a basis for price; (2) *Tough Ingot*, which is generally used by founders for general purposes, and branded with some well-known smelter's name or mark; (3) *Best Selected*, is purer and of more uniform quality, and used for high class work. *Electrolitic* is the purest form, however, for highly important work.

In its simple state it is employed chiefly for pipes on account of its ductility and strength, and in some measure because it can be joined by brazing, so as to be as strong there as the original sheet, and for plates or sheets with which to make the fire-boxes of the locomotive boiler. It does not generally corrode under the action of sea-water or air, but does sometimes waste by the mechanical action of water and steam moving at high velocities over its surface. In some few localities the water seems to have a destructive effect on this metal, owing no doubt to the presence of free gases mechanically mixed with it; of these gases, sulphuretted hydrogen is the worst in the water of rivers and ports, and chlorine in sea-water; the presence of bromine, too, is often detected by the smell so often noticeable in the hot-wells of some engines.

Copper pipes of large size (6 inches and upwards) are always made from sheets, curved into the required form by rolling or hammering, and brazed at the seams. Copper pipes can be made seamless, practically of any size, by electro-deposits, etc., on Cowper-Coles method of treatment, but steel is now superseding this metal.

Smaller pipes are sometimes made in the same way, but generally by "drawing." The feed, blow-off, and scum pipes in the Navy are always of solid-drawn copper, because there should be no seam in such important pipes; and the main steam pipes were also solid drawn when made of copper, and not exceeding 6 inches diameter, but now they are of solid-drawn steel.

Solid-drawn pipes are very seldom used in the mercantile marine, partly because they are somewhat more expensive, but chiefly because they were not so uniform in thickness as the brazed ones, and thought to be more liable to split unless carefully manufactured from very soft tough copper.

The strength of copper depends somewhat on its purity, but principally on the amount of work it has undergone, especially in the cold state. Copper castings have an ultimate strength of only about 10 tons per square inch; when forged its strength is increased to about 15 tons, and when rolled into bars is 16 tons; if a small proportion of phosphorus is added (about 2 per cent.) the strength is increased to 20 tons; pure copper when drawn out into wire has a strength of about 28 tons before, and 18 tons after annealing. Sheet copper has an average strength of about $13\frac{1}{2}$ tons, and for purposes of calculation may be assumed to be 30,000 lbs. per square inch. Small quantities of arsenic and some other metals improve copper very much. With only $\frac{1}{2}$ per cent. of arsenic plates have an ultimate strength of 33,420 lbs., with a stretch of 37 per cent. (Roberts-Austen). Wire made of copper with 1 per cent. of aluminium stood 78,000 lbs. per square inch, and French wire made of copper with a small addition of silicon 128,000 lbs. Wire with 0·529 per cent. of antimony stood 78,000 lbs. as against 49,000 lbs. with pure copper wire.

The specific gravity of sheet copper is 8·805; the weight of a cubic foot is 550 lbs., that of a cubic inch 0·318 of a pound, and that of a square foot 1 inch thick 45·83 lbs.

Tin.—This metal, although seldom used alone, is very important, as forming one of the chief constituents of bronze or gun-metal, but now more so as being the chief ingredient of some white metals. The best qualities are obtained from Cornwall, and the chief supply of this metal was from that country; of late years, however, considerable quantities have been imported from the Dutch East Indies, Malacca, and Australia, and although not so pure as the Cornish tin, the price of the latter has been very considerably affected by the supply.

Tin is used as a protective covering to other metals on account of its immunity from the corrosive action of salts and acids. The Admiralty formerly required all condenser tubes to be coated with tin when fitted in iron condensers; and this practice was also followed by some Mercantile Shipping Companies. Since sea-water ceased to be used as feed supply the tinning of condenser tubes has gone out of practice. Sheet tin, which is thin sheet iron or steel coated with tin, is used for "liners" between "brasses," as well as for making oil feeders, lamps, cups, etc.

Tin mixed with small quantities of copper, antimony, etc., is used under the name of white metal to line and face bearings.

The tensile strength of tin is too low to admit of its being used alone in construction; its ultimate strength when cast is only 2·11 tons per square inch. Its specific gravity is about 7·3, consequently the weight of a cubic foot is 456 lbs., and that of a cubic inch 0·264 lb.

Zinc or Spelter.—This metal also is seldom used alone, but is very largely employed to alloy with copper to form brass, and with it and some other metals in small quantities the well-known strong zinc bronzes. The best kinds come from Australia and the Continent; the Silesian spelter is the purest, and generally used in making brass for rolling into sheets or drawing into tubes and rods. Ordinary zinc contains lead in appreciable quantities. *Electrolitic* is quite pure, and is best for high-class alloys.

In its simple state zinc is used by marine engineers to prevent corrosion in boilers, condensers, and hot-wells, and to protect the ship's plating near the bronze propellers. Cast blocks, or, better still, pieces of rolled bar or sheet of this metal are placed in metallic contact with the iron of the boiler in such places as have been found by experience to require protection. The purer the zinc is, the more perfect is the protection afforded; but unless there are exceptional circumstances affecting the feed-water, common zinc or even "hard spelter" (residuum from the galvanising bath) will form a sufficiently strong galvanic couple to prevent deterioration of the iron surfaces.

Zinc is also employed as a covering for iron or steel, to protect it from the action of sea-water, etc., and being much cheaper than tin, and easily applied to the surface of the iron, is used on a far more extended scale than tin. Zinc is also used as the principal constituent of certain kinds of white metal made for bushes working in water.

The tensile strength of zinc is even lower than that of tin, being only 1·336 tons per square inch when cast. Its specific gravity is 7·0, consequently the weight of a cubic foot is 437 lbs., and that of a cubic inch 0·253 lb.

Lead is nearly always used alone, and the purer it is, the more valuable for engineering purposes. In the mercantile marine the bilge piping is often made of lead, and the pipes for emptying and filling the ballast tanks are also sometimes of lead. It is used for these purposes because of its resisting the corrosive action of sea and bilge water, and being very ductile it can be easily bent to follow the curves and corners of the ship's bottom.

Sheet lead is sometimes used to protect the covering of boilers from wet, and also to cover engine-room floors when made of wood, as it gives a better foot-hold than iron plates. It is employed, too, for jointing pipes, etc., when the flanges are rough and uneven. Of late years it has been largely used as the chief constituent of certain white bearing metals, hardened with antimony.

It is sometimes used by moulders to give a good colour to common brass for ornamental purposes, and to cause it to be readily turned in the lathe; but even the smallest addition of this metal tends to reduce the strength of brass, and it should be, therefore, generally avoided.

The tensile strength of sheet lead is only 0·81 ton per square inch, and that of lead pipe is 1 ton. The specific gravity is 11·418, consequently the weight of a cubic foot is 712 lbs., of a cubic inch 0·412 lb., and of a square foot 1 inch thick, 59·3 lbs.

Aluminium.*—This metal, formerly so rare and very expensive, is doubtless destined to become a very important one to marine engineers from its extreme lightness as it has to the automobile engineer. Its price when introduced (about £1,000 per ton) quite precluded its use, even to form an alloy; in normal times, when it can be purchased at £80, engineers can use it freely to make bronze, which is as strong and elastic as mild steel. Aluminium, with the addition of 6 per cent. of copper, rolled into sheets and bars having a fair strength is still light, and cost only 1s. per lb. Such sheets have an ultimate strength of 11 to 12 tons annealed, and 14 to 16 tons unannealed, as against 7 to 8 tons in the case of sheets of pure aluminium. Aluminium bronzes in the cast state are very porous, and the crystalline structure very coarse. A little aluminium improves all zinc bronzes, but damages tin bronzes.

In the pure state when drawn into wire, it has a tensile strength of about 8 tons per square inch; by hammering cold, the strength is raised from about 7 tons (as cast) to about 12 tons. An alloy of 90 per cent. of copper and 10 per cent. of aluminium, when rolled, has an ultimate tensile strength of 32 to 40 tons per square inch, and Professor Arnold found that an alloy of 92·5 of copper and 7·35 of aluminium registered the highest resistance to alternating stresses he had ever observed—viz., 1,359, as against 657 of the 9·9 per cent. mixture.

The specific gravity of aluminium is only 2·56, consequently a cubic foot weighs 160 lbs., a cubic inch 0·092 lb., and a square foot 1 inch thick, 13·33 lbs.

Duralumin is a good alloy made and sold by Vickers, Ltd., at such a moderate cost as ensured its general use in engineering practice, considering its great strength and lightness. It consists of about 90 per cent. of aluminium, and has a specific gravity of 2·8, and, therefore, very little heavier than pure aluminium. Its melting point is about 1,200° F.

* The standard price in 1918 is £230 per ton.

Its tensile strength may be as high as 40 tons per square inch, with an extension, however, somewhat small; but at 28 to 30 tons it is quite satisfactory, being as much as 15 per cent. in 2 inches, and with a tensile strength of 25 tons 20 per cent. elongation is obtained. It is not used in the cast state, but was supplied in sheets from 24 to 28 inches wide, and up to 10 S.W.G. thick at 2s. 4d. per lb., which corresponds to 9d. for an equal quantity (volume) of copper or brass, or to 10d. for steel. It was also rolled exceedingly thin in narrower widths for 2s. per pound. Hot-rolled bars from $1\frac{1}{16}$ to 3 inches diameter cost only 2s. 1d., and was supplied down to $\frac{3}{8}$ inch diameter at 2s. 4d. per pound. During the war the price is more than doubled.

Cold-drawn bars were only $1\frac{1}{2}$d. per pound dearer. This metal can be worked hot or cold, and, therefore, made into tubes, sectional bars, etc., stampings, and forgings. It is non-magnetic, is little affected by sea or fresh water or damp air, and takes a good polish.

Antimony is used in very small quantities, to harden other metals and alloys, especially the white metals for bearings.

Alloys.—Strictly speaking, alloys of copper and zinc only can be called *brass*, but ordinary bronze, as made for bearings, liners, and bushes, is often called brass, and from these circumstances the liners of journals and pin-bearings are called "brasses." Alloys of copper and tin, or those of copper and tin together with zinc or other metal, are called *bronze.* Alloys of copper and zinc with manganese or other modifier are now generally called *zinc bronzes.*

Brass.—The *yellow brass* used for ornamental castings is usually composed of two parts of copper and one part of zinc; when carefully made, castings of yellow brass have a tensile strength of 12 to 13 tons, but the ordinary yellow brass, as supplied to founders, has a strength of only 10 to 11 tons; it is fairly tough, but too soft for general purposes.

Muntz's Metal, composed of three parts of copper and two of zinc, can be rolled out into bars and sheets, so as to have an average tensile strength of 22 tons per square inch, and in some cases bars of this metal have a strength as high as 27 tons. It is very ductile, and can be forged when hot; it will stretch very considerably before fracture, and may be used for springs when hammered or cold rolled, and not annealed.

Its specific gravity is 8·2, consequently a cubic foot weighs 512 lbs., that of a cubic inch 0·296 lb., and that of a square foot 1 inch thick is 42·7 lbs.

Naval Brass.—By the addition of 1 per cent. of tin to Muntz's metal, it has a better power of resisting the action of sea-water, while retaining all its other properties. An alloy of 62 copper, 37 zinc, and 1 tin is known as *naval* brass, because of its use originally in the construction of naval composite ships, and for the bolts of the engine-fittings which are exposed to sea-water in warships. This metal can be forged hot, and bent cold two double; its strength is superior to the ordinary Muntz's metal, and some specimens rolled cold and unannealed have been proved to have an ultimate strength of nearly 40 tons per square inch. As usually supplied, it has an ultimate strength of 27 tons, with an elongation of 19 to 20 per cent. in inches; the elastic limit is about 19 tons.

Brass Tube Metal.—The ordinary brass condenser-tubes are made of a composition containing 70 per cent. of copper and 30 per cent. of zinc, but the Admiralty require them to be composed of 70 per cent. of best selected copper, 29 per cent. of Silesian zinc, and 1 per cent. of tin,* and all tubes supplied to the Navy have to undergo the test described on p. 363. The strength of the metal of tubes made with 70 per cent. of best selected copper and 30 per cent. Silesian zinc is as high as 36 tons per square inch.

Gun-metal or Bronze.—There is no particular mixture to which this name belongs, as it is applied promiscuously to any composition of copper and tin, or copper, tin, and zinc.

The best known composition, and one which has high strength, is fairly hard, and very tough, is that containing 90 per cent. of copper and 10 per cent. of tin. Its tensile strength, when carefully made, is 17 tons per square inch; its specific gravity is 8·66, consequently a cubic foot weighs 561 lbs., a cubic inch 0·325 lb., and a square foot 1 inch thick 46·8 lbs. To insure sound castings, however, it is necessary to add a small quantity of zinc.

A much harder metal is made by mixing 84 per cent. of copper with 16 per cent. of tin; its tensile strength is 16 tons, specific gravity 8·56, and the weight of a cubic foot is 534 lbs.

For heavy bearings, where hardness is of more importance than strength, although the metal must not by any means lack strength, a good metal is made by mixing 79 per cent. of copper with 21 per cent. of tin: its tensile strength is nearly 14 tons when carefully made, and the average is 13½ tons; the specific gravity is 8·73, and the weight of a cubic foot is 544 lbs.

Admiralty Bronze of 88 copper, 10 tin, and 2 zinc best quality has an ultimate tensile strength of 20 tons, with an elongation in 2 inches of 25 per cent.; when of good commercial metals test bars should give 18·5 tons with 20 per cent., and castings stand at least 16 tons with 15 per cent. A little phosphorus in the tin or copper improves the strength generally. With 0·5 of lead added, the tensile strength at temperatures up to 550° Fah. is maintained.

Phosphor-Bronze.—This metal is composed of copper and tin, with a small proportion of phosphorus. It is harder than the ordinary bronze, very close grained, and of superior strength. The average ultimate strength is about 17 tons per square inch, while that of some specimens is as high as 22 tons. The singular thing about this metal is its high elastic limit, being very nearly that of the ultimate strength. Its resistance to shearing in the cast state is also exceptionally high. Great care is required in melting, and repeated meltings very much reduce its virtue. It may be rolled out into extremely thin sheets, or drawn into wire, when the average tensile strength is 56 tons per square inch. Phosphor bronze sheet is used for the valves of air-pumps (*vide* p. 393).

This metal is also used for bearings, brasses, propeller-blades and bosses, pump-rods, etc.

The Original Manganese Bronze is now seldom or never used. It contained considerable quantities of manganese, had a dull brown fracture, but a high tensile strength. The present alloy is good zinc bronze improved by the addition of ferro-manganese. The manganese is said to deoxidise any copper oxides which may be mechanically mixed with the copper, so "rendering the metal more dense and homogeneous." The No. 1 quality, which was used for forgings and rolling into rods, plates, sheets, angles, etc., when cast in metal moulds had an ultimate strength of 24 tons, and an elastic limit of 14 tons per square inch.

* Some makers use 2 per cent. of lead instead of the tin, and claim as good or better results. Bemal metal, which is a specially made 70/30 mixture, resists corrosion better still.

Rolled rods, plates, etc., as now made by the Company, have, when mild, an ultimate strength of 28 tons, and an elongation of 40 per cent.; but when so required it can be made with an ultimate strength of 30 to 32 tons, an elastic limit of 15 to 17 tons, and an ultimate elongation of 15 to 20 per cent.

By cold rolling the strength can be raised to even 40 tons, but the elongation is then reduced to 10 per cent.

The Manganese Bronze Company manufacture various other qualities of bronze for special purposes, such as follows :—

Crotarite is designed for such things as boiler stays and other fittings as are exposed to heat and required to sustain heavy stresses. This alloy has a melting point almost as high as copper and a coefficient of expansion no greater. It contains no zinc, is highly malleable, and can be riveted cold. Its ultimate tensile strength is 25 to 26 tons per square inch, with an elongation in 2 inches of 30 to 40 per cent. Its elastic limit is high, being no less than 16 tons.

Immadium is another special alloy suitable for shafts, spindles, and rods of pumps of all kinds, inasmuch as it is practically incorrodible in sea-water, and is even unaffected by water containing small quantities of acids, while its tensile strength is exceedingly high. The strongest kind has an ultimate strength of 40 tons per square inch, with an elongation of 20 to 25 per cent. in 2 inches, and an elastic limit of 20 tons per square inch. The mild variety has an elastic limit of 18 tons, and an ultimate tensile strength of 36 tons per square inch, with an elongation of 25 to 30 per cent.

This alloy is valuable for purposes such as boiler mountings and fittings, cylinder fittings, and connections, inasmuch as it maintains a high tensile strength when heated to quite high temperatures, for at 500° the elastic limit is 9 to 10 tons per square inch, while the ultimate tensile strength is 23·5 tons, with an elongation of 22·5 per cent., while at 420° F. the temperature of steam of 315 lbs. pressure absolute the elastic limit is as high as $8\frac{1}{2}$ tons, and the ultimate tensile strength 24·5 tons per square inch, with an elongation of 25 per cent. in 2 inches. Castings of this material are usually sound and easily machined, so that they may be substituted for steel ones with advantage, seeing that they are even stronger than steel and no more costly, if risk of wasters and cost of machinery are taken into account.

Turbadium is a good alloy patented by the Parsons Manganese Bronze Company, and supplied for the casting of propeller blades. It is a zinc bronze consisting of about 55 per cent. of copper, 43 of zinc, 1·44 of iron, and small quantities of manganese and tin. The important point, however, is that the structure of this alloy as made is essentially β or polyhedric in character, so as to withstand erosion from which the surface of all bronze propellers suffer when run at very high rates of revolution as in turbine steamers. Propellers of this material have been supplied by the Company to the steamships "Lusitania," "Mauritania," etc., with most satisfactory results. Fig. 178 is a photo of one of those fitted to the "Mauritania," taken after many months of service in the Atlantic; it shows no signs of erosion, as did those of other bronze almost immediately after going on service.

The elastic limit of this material is 18 tons, and the ultimate tensile strength 40 tons with an elongation of 18·58 per cent. in 8 inches. It can be treated by press or hammer at a cherry-red heat, so that all buckles and bends due to striking quays or wreckage can be removed and the blade restored to proper shape and pitch quite easily.

Stone's Bronze is a zinc and copper composition, possessing all the characteristics of modern manganese bronze, and is used very extensively for propeller blades and engine castings, where high tensile strength and great toughness is necessary.

Bull's Metal is another of the zinc bronzes with either nickel or ferro-manganese added, having a high tensile strength with great toughness; in the rolled state it has resistance to compression as high as 100 tons per square inch.

Melloid is a tin bronze with a small percentage of phosphorus. It is made by Bull's Company, and can be rolled and forged hot, and rolled cold.

Delta Metal.—The invention of Mr. Dick is a high-class bronze, somewhat similar to manganese bronze, but rather harder and with a higher tensile strength. Castings made from it have an ultimate strength of 34 tons, and resist torsion better than the other zinc bronzes, which are, as a rule, disappointing in this respect.

Parsons' White Brass.—This is a most valuable material for facing and lining bearings; the composition of No. 2 quality is tin, 68; zinc, 30·5; copper, 1; and lead, 0·5; its success as a bearing metal is most unqualified, and it does well in crank-pin brasses at moderate speeds.

Babbit's White Metal.—This was, for very many years, almost the only white metal used for bearings, and until modern times was without a rival. It is composed of 10 parts of tin, 1 of copper, and 1 of antimony.

Admiralty White Metal.—A very good white metal is made by mixing 6 parts of tin with 1 of copper, 6 parts of tin with 1 of antimony, and adding the two mixtures together, and is used in all Admiralty work.

Plumtine is a comparatively cheap and very efficient white metal, composed of 48·5 per cent. of lead, 40·5 of tin, 11 of antimony, and 0·5 of copper.

Magnolia is also a white metal consisting largely of lead, and is well known as an excellent bearing metal.

Fenton's White Metal, which is used for stern bushes and the bushes of paddle-wheels, etc., is composed of 8 parts of zinc, 1·66 of tin, and 0·44 of copper; it is fairly tough and hard, and in sandy water resists wear exceedingly well.

Stone's White Bronze is also an excellent metal for bearings and crank-pin brasses, especially of heavy engines running at high speed.

German Silver.—An alloy of nickel and copper and zinc, known by this name, is used largely for instruments, and for domestic implements after being electro-plated. It is of yellowish-white appearance, fairly hard and tough, and capable of being worked cold.

Richard's Plastic Metal.—This is a white metal, which melts at a moderately low temperature, and can be worked with a soldering-iron. It is very useful, therefore, for mending bearings and for coating damaged brasses; it also does for a filling metal, working well in bearings and guides.

TABLE XCIV.—Composition of Metals and Alloys.

Metal or Alloy.	Ultimate Tensile.	Extension.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
		Per cent.													
Pig iron, good grey,	...	...	95·0	...	2·99	...	0·97	...	...	...	...	...	...	...	0·50
,, common grey,	...	...	94	...	3·44	0·43	1·13	...	...	...	...	...	...	...	1·24
,, hæmatite,	...	...	95·7	...	3·27	0·37	0·67	...	...	...	...	...	...	...	0·28
Steel, mild Siemens,	58,000	25	99·1	...	0·15	0·55	0·02	...	...	...	...	...	...	...	0·05
,, high tensile Siemens,	80,000	22	...	...	0·40	...	...	...	...	...	...	...	...	...	...
,, for tools, ordinary,	124,000	5	...	...	1·00	...	...	...	...	...	...	...	...	...	...
Nickel steel, mild forging,	85,000	30	95·5	3·25	·285	·687	·082	...	...	...	...	...	...	...	0·04
,, strong ,,	198,000	12	88·0	11·39	0·20	...	...	...	...	...	...	...	...	...	...
Manganese steel (Hadfield),	130,000	35	...	...	...	11·0	...	...	...	...	...	...	...	...	...
Copper plates, strong,	33,400	37	...	...	...	...	...	99·5	...	...	...	...	0·50	...	...
,, wire,	78,000	...	...	...	...	...	...	99·0	...	...	1·00	...	...	...	...
,, ,,	128,000	...	...	...	...	...	trace	97·1	1·14	1·12	...	...	...	...	...
Muntz metal, rolled,	56,000	27	...	...	...	...	...	60·0	...	40·0	...	...	...	...	...
Naval brass, ,,	60,800	19	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
,, condenser tubes,	...	...	...	...	...	...	...	70·0	1·00	29·0	...	...	...	...	...
,, ,, plates,	...	...	...	...	...	...	...	62·0	1·00	37·0	...	...	...	...	...
Aluminium sheets,	34,400	3·7	...	...	...	...	...	6·00	...	...	94·0	...	...	...	...
,, bronze, cast,	60,400	25·6	...	...	...	...	2·00	90·5	...	...	7·5	...	...	...	...
,, ,, ,,	56,000	42·0	...	...	...	...	...	95·0	...	...	5·0	...	...	...	...
Phosphor bronze (wheel), cast,	39,200	...	...	...	...	...	...	82·2	12·95	...	...	...	...	4·28	0·52
Manganese ,, (Parsons), ,,	67,000	21·5	1·67	...	0·06	16·86	...	81·0	...	...	...	...	...	...	...
Turbadium ,, ,, ,,	90,200	18·6	1·44	...	...	0·13	...	54·6	0·70	43·13	...	...	...	...	...

TABLE XCIV.—COMPOSITION OF METALS AND ALLOYS (*Continued*).

Metal or Alloy.	Ultimate Tensile.	Extension.	Iron. Fe.	Nickel. Ni.	Carbon. C.	Manganese. Mn.	Silicon. Si.	Copper. Cu.	Tin. Sn.	Zinc. Zn.	Aluminium. Al.	Antimony. Sb.	Arsenic. As.	Lead. Pb.	Phosphorus. Ph.
		Per cent.													
Stone's bronze, cast,	77,500	...	1·67	...	...	...	...	56·1	1·05	40·6	...	...	...	0·47	...
Bull's metal, rolled,	78,000	34	...	...	...	...	...	57·1	1·49	40·2	0·15	...	...	0·34	...
Delta ,, ,,	89,000	...	0·69	...	...	2·06	...	57·4	...	38·9	...	...	...	0 71	...
Roma ,, ,,	77,500	...	0·03	...	...	...	...	58·6	..	40·7	0·20	...	...	0·39	...
Aich's ,, ,,	57,300	25·6	1·50	...	...	...	...	59·8	0·80	37·9	...	...	...	...	...
Sterro ,, ,,	85,000	...	1·50	...	...	...	...	60·0	...	38·1	...	...	...	...	...
German silver (English),	...	...	...	19·1	...	...	...	61·3	...	19·1	...	...	...	...	...
Argentan (Basileus),	...	...	...	26·0	...	...	...	52·0	...	22·0	...	...	...	...	...
Gunmetal,	38,000	...	...	...	...	...	...	91·0	9·0	...	...	...	...	...	...
Admiralty bronze,	30,000	...	...	...	...	...	...	87·0	8·0	5·0	...	...	...	...	...
,, ,, for fittings,	...	...	...	...	...	...	...	87·0	11·0	2·0	...	...	...	...	...
,, white metal,	...	...	...	...	...	...	...	7·15	85·7	...	...	7·15	...	...	...
Babbit's white metal,	...	...	...	...	...	...	...	8·33	83·3	...	...	8·33	...	...	...
Parsons' white brass, No. 2,	...	...	...	...	...	...	...	1·00	68·0	30·5	...	...	...	0·50	...
Magnolia metal,	...	...	...	...	...	...	...	...	4·60	...	...	13 2	...	82·0	...
Plumtine ,,	...	...	...	...	...	...	...	0·50	40·5	...	...	11·0	...	48·5	...
Fenton's white metal,	...	...	...	...	...	...	...	4·40	16 6	79·0	...	...	...	...	...
Bell metal,	...	...	...	...	...	...	...	76·5	23·5	...	...	...	...	...	...
Britannia metal,	...	...	...	...	...	...	...	1·46	90·6	...	...	7·81	...	...	...
Solder, fine,	...	...	...	...	...	...	...	...	66·6	...	...	...	...	33·4	...
,, plumber's,	...	...	...	...	...	...	...	...	33·3	...	...	...	...	66·7	...

TABLE XCV.—COMPOSITION OF WHITE (BEARING) METALS.

Name of Metal.	Cu.	Sn.	Sb.	Zn.	Pb.	Fe.	Miscellaneous.
Dewrance's Locomotive,	22·2	33·3	44·4	...	...	...	...
For gland packings,	7·8	88·1	3·5	...	2·6	...	...
Used in the German navy,	7·5	85·0	7·5	...	...	...	...
,, French ,,	7·0	7·5	...	78·5	7·0	...	...
,, British ,,	5·5	86·0	8·5	...	...	...	...
Fenton's,	4·4	16·6	...	79·0	...	...	...
Magnolia,	...	...	21·0	...	78·0	1·0	...
,,	...	4·6	13·0	...	82·0	...	...
Kingston's,	6·0	88·0	...	...	...	...	6 Hg
Parsons' white brass,	5·6	17·5	0·8	76·1	...	...	...
,, ,, metal,	...	58·5	2·0	39·5	...	...	...
,, ,, ,,	1·0	68·0	...	30·5	0·5	...	...
For common bearings,	10·0	...	10·0	...	80·0	...	...
,, heavily loaded bearings,	64·0	5·0	...	...	30·0	...	1 Ni
Plumtine,	0·5	40·5	11·0	...	48·5	...	...

TABLE XCVI.—Properties of Metals.

Metal or Alloy.	Ultimate Strength.	Extension.	Elastic Limit.	Compressed Strength.	Shear.	Specific Gravity.	Weight of Cubic Inch.	Weight of Sq. foot 1 inch Thick.	Melts at ° F.	Conducts Heat. Silver 100	Extension per 10° F.
	Lbs.	Per cent.	Lbs.	Lbs.	Per cent.						
Cast iron, good castings,	25,000	...	...	110,000	162·0	7·20	0·260	37·50	2100	...	...
Wrought iron, good boiler quality,	50,000	...	26,000	...	80·0	7·68	0·280	40·0	2800	11·90	·0000670
,, ,, bar ,,	60,000	...	30,000	...	...	7·62	0·276	39·70	,,	,,	,,
,, ship plate ,,	45,000	...	27,500	...	...	7·68	0·280	40·0	,,	,,	,,
Steel castings, good mild,	63,000	...	34,000	...	...	...	...	...	2500	,,	,,
,, wrought (0·15 carbon),	58,000	25·0	29,000	...	80·0	7·86	0·284	40·94	,,	,,	,,
,, ,, (0·40 ,,),	78,000	22·0	33,000	...	...	...	...	...	...	...	...
,, ,, (,, ,,) oil tempered,	90,000	...	55,000	...	...	...	...	...	...	...	...
Nickel steel (3¼ Ni, 0·29 C), forged,	85,000	30·0	54,000	...	...	...	...	...	...	...	...
,, (3¾ Ni), oil tempered,	90,000	...	60,000	...	...	...	...	...	...	...	...
,, mild quality boiler plate,	74,000	...	43,000	...	...	...	...	...	...	...	...
,, high tensile ,,	114,000	...	64,000	...	...	...	...	...	...	...	...
Chrome steel (2·5 Cr, 0·39 C), forged,	65,400	...	52,600	...	...	...	...	...	...	...	...
,, (,,), unannealed,	120,900	...	97,400	...	...	...	...	...	...	...	...
Vanadium steel (0·25 per cent. V),	88,000	24·0	76,300	...	...	...	...	...	...	...	...
Chrome vanadium steel (1·0 Cr, 0·15 V),	117,800	25·0	77,000	...	77·2	...	...	...	...	...	...
Manganese steel (11·0 Mn), forged,	134,000	35·0	...	...	...	...	...	...	...	...	...
Nickel, cast,	36,400	...	24,600	...	...	8·30	...	...	2820	...	...
,, forged,	72,100	...	25,500	...	...	8·80	0·318	45·80	2820	...	·0000705
German silver (19 Ni), rolled,	...	...	...	...	...	8·80	0·318	45·80	...	...	...
Copper, annealed sheets,	30,000	...	5,600	...	86·7	8·85	0·320	46·10	2000	73·6	·0000930
,, bars,	36,000	...	...	...	...	,,	...	...	,,	,,	,,
Zinc, cast,	3,000	...	...	...	...	7·00	0·253	36·42	800	28·1	·0001620
,, rolled,	7,500	...	...	...	...	,,	...	...	,,	,,	,,
Tin, cast,	4,700	...	...	...	...	7·30	0·264	38·00	440	15·2	·0001280
Muntz metal, rolled,	56,000	27·0	36,000	...	...	8·20	0·296	42·72	1630	...	...
Naval brass, ,,	60,800	19·0	42,400	...	55·8	8·20	0·296	42·72	1640	...	...

TABLE XCVI.—PROPERTIES OF METALS (*Continued*).

Metal or Alloy.	Ultimate Strength.	Extension.	Elastic Limit.	Compressed Strength.	Sheer.	Specific Gravity.	Weight of Cubic Inch.	Weight of Sq. Foot 1 Inch Thick.	Melts at ° F.	Conducts Heat.	Extension per 10° F.
	Lbs.	Per cent.	Lbs.	Lbs.	Per cent.						
Manganese bronze (Parsons), cast No. 2,	67,000	24·5	33,000	..	..	8·51	0·308	44·34	1,660	..	..
,, ,, rolled, annealed,	67,000	26·2	33,000	..	..	8·40	0·304	43·76	,,	..	..
Stone's bronze, cast,	77,500	..	38,200	127,000	..	..	..	..	..	..	..
,, rolled,	80,600	..	66,000	..	..	..	..	..	..	..	..
Delta metal, rolled,	89,000	..	..	..	..	8·40	0·304	43·76	1,800	..	..
Bull's metal, cast,	70,000	..	29,600	176,000	..	8·30	0·300	43·24	1,800	..	..
,, rolled,	78,000	34·0	60,000	224,000	..	,,	,,	,,	..	..	..
Aitch's metal, rolled,	57,300	25·6	..	..	..	..	..	..	..	..	..
Phospher bronze (wheel B), cast,	39,200	..	38,900	..	113·0	8·60	0·311	44·80	1,900	..	..
,, ,, rolled,	68,500	..	62,000	..	63·0	,,	,,	,,	,,	..	..
Silicium bronze wire,	105,000	..	..	..	..	..	..	..	..	..	..
Admiralty bronze, cast,	30,000	..	17,000	..	..	8·66	0·325	46·80	1,700	..	..
Aluminium (pure comml.), cast,	11,800	..	..	..	..	2·56	0·092	13·33	1,160	31·33	·0000280
,, (,,), bar rolled,	28,200	..	19,300	..	..	2·70	0·098	14·06	,,	,,	,,
,, (,,), plates, rolled	16,500	..	9,600	..	..	2·70	0·098	14·06	,,	,,	,,
,, (6·0 Cu), plates rolled,	34,400	3·7	30,600	..	..	3·07	0·111	16·00	..	..	..
Aluminium bronze (10 Al), rolled,	100,000	22·0	..	..	..	7·65	0·277	39·86	1,750	..	..
,, (5 Al), cast,	56,000	50·0	..	..	..	..	..	..	..	..	..
Antimony, cast,	..	..	..	..	..	6·71	0·243	35·00	1,166	4·03	..
Bismuth, cast,	..	..	..	..	..	9·80	0·355	51·05	514	1·80	..
Lead, sheet,	3,300	..	..	..	..	11·37	0·411	59·24	620	8·50	..
Manganese, cast,	..	..	..	..	..	8·01	0·289	41·52	3,450	..	..
Platinum, rolled,	..	..	..	..	..	21·50	0·778	112·0	3,225	8·40	..
Silver, rolled,	..	..	..	..	..	10·53	0·381	54·86	1,730	100·0	.
Parsons' propeller bronze (average),	77,600	26·93	..	..	..	..	..	..	..	..	..
,, immadium bronze, cast,	80,600	15·0	36,100	..	..	..	..	..	—	..	.
,, ,, forged,	90,000	20·0	45,000	..	..	..	..	..	..	..	..
,, ,, ,,	80,600	25·0	40,300	..	..	..	..	..	..	..	..
,, turbadium bronze, special propeller cast,	90,200	18·58	40,320	..	..	..	..	..	..	..	..

TABLE XCVII.—Prices of Materials.*

Materials at Maker's Works.	Prices, 1912.	Prices, 1906.
Pig iron, Blaenavon, cold blast, No. 3.	None.	£9 2 6 per ton.
,, Shropshire, ,, ,,	£6 0 0	5 12 6 ,,
,, Hæmatite, West Coast, ,,	3 13 6	3 5 0 ,,
,, ,, East Coast, ,,	3 11 6	3 8 6 ,,
,, Scotch, good brand, ,,	3 6 0 (1917, £7 15s.)	3 2 6 ,,
,, ,, No. 1,	3 11 0	3 8 0 ,,
,, Cleveland, ,, No. 3.	2 14 3	2 9 0 ,,
,, ,, ,, No. 1,	2 19 6	2 10 0 ,,
,, Lincoln, ,, Foundry	3 0 6	2 18 0 ,,
,, Derby, ,, ,,	3 2 6	3 1 6 ,,
Wrought Iron, Cleveland bars, ordinary,	7 15 0	7 5 0 ,,
,, ,, ,, best,	8 5 0	7 15 0 ,,
,, ,, ship plates,	8 10 0	7 5 0 ,,
,, ,, ,, angles,	7 15 0	7 5 0 ,,
,, Staffordshire, best bar,	8 10 0	8 0 0 ,,
,, sheets,	8 5 0	8 5 0 ,,
Steel, angles and tees, ship quality,	7 7 6	6 12 3 ,,
,, plates, ship quality,	7 15 0	7 0 0 ,,
,, boiler plates, Board of Trade, etc.,	8 15 0	8 0 0 ,,
,, forgings, plain shafts, with couplings,	£15 to 20 0 0	£14 to 18 0 0 ,,
Copper ingots G.M.B.,	70 0 0	79 5 0 ,,
,, ,, tough,	74 10 0 (1917, £136)	84 5 0 ,,
,, ,, best selected,	74 10 0	84 10 0 ,,
,, locomotive plates,	84 10 0	(basis) 94 10 0 ,,
,, sheets,	82 0 0	,, 92 0 0 ,,
,, pipes, solid drawn, $1\frac{1}{4}$ in. to 3 in.,	0 0 $10\frac{1}{4}$	,, 0 0 $10\frac{3}{4}$ per lb.
,, ,, brazed,	0 0 10	,, 0 0 $10\frac{1}{2}$,,
Brass condenser plates, Mercantile,	0 0 $7\frac{1}{2}$	0 0 $7\frac{7}{8}$,,
,, ,, ,, Naval,	0 0 $7\frac{3}{4}$	0 0 $8\frac{1}{8}$,,
,, tubes, 66 per cent copper,	0 0 $8\frac{5}{8}$	0 0 9 ,,
,, ,, ,, 70 ,, ,, Mercantile,	0 0 $8\frac{3}{4}$	0 0 $9\frac{1}{8}$,,
,, ,, ,, 70 ,, ,, Admiralty,	0 0 $9\frac{1}{4}$	0 0 $9\frac{1}{2}$,,
Tin, English ingots,	205 0 0 (1917, £300)	166 0 0 per ton.
Zinc slabs, Silesian,	25 15 0	25 2 6 ,,
,, ,, best,	26 15 0 (1917, £52)	25 12 6 ,,
Aluminium, ingots, Nos. 1, 2, and 4,	84 0 0 (1917, £230)	160 0 0 ,,
,, ,, No. 6,	76 0 0	150 0 0 ,,
,, sheets, from Nos. 1 to 30, wire gauge,	105 14 0	£150 to 180 0 0 ,,
Lead, pig,	19 15 9 (1917, £32)	17 15 0 ,,
,, pipe,	21 15 0	19 10 0 ,,
Nickel ingots, of good quality,	0 1 0	0 1 0 per lb.
,, refined (98 to 99 per cent. purity)	0 1 6	..
German silver sheet,	1s. 3d. to 0 1 4	1s. 2d. to 0 1 5 ,,
Muntz metal sheet,	0 0 $8\frac{1}{2}$	0 0 $7\frac{3}{4}$,,
Naval brass rods,	0 0 $8\frac{3}{4}$	0 0 $8\frac{1}{4}$,,
Phosphor bronze, ingots, for general castings,	106 0 0	102 0 0 per ton.
,, ,, suitable for bearings,	116 0 0	113 0 0 ,,
,, rods,	0 1 $0\frac{3}{4}$	(basis) 0 1 $2\frac{1}{2}$ per lb.
,, sheet,	0 1 $2\frac{3}{4}$	,, 0 1 4 ,,
Manganese bronze, ingots,	..	£90 to 105 0 0 per ton.
Antimony, cakes,	27 10 0 (1917, £90)	68 0 0 ,,
Bismuth,	0 10 0	0 7 0 per lb.
Platinum wire,	9 5 0	4 11 6 per oz. Troy
,, sheets,	9 5 0	4 11 6 ,,

* Since August, 1914, the prices of all metals and alloys have been abnormal, and generally more or less artificial.

TABLE XCVIII.—STANDARD TESTS OF STEEL AS ADOPTED IN U.S. AMERICA.

	Specification for	Tensile Strength.	Yield Point.	Elongation (c).	Contraction of Area.	Bending Test. D = Diameter. T = Thickness.
		Lbs. per Square Inch.	Lbs. per Square Inch.	Per cent.		
Steel castings,	Ordinary castings,	...	...	...	...	...
	Tested castings, hard,	85,000	38,250	15	20	...
	,, medium,	70,000	31,500	18	25	90°, D = 2 T.
	,, soft,	60,000	27,000	22	30	120°, D = 2 T.
,, axles,	Car, engine truck, and tender truck,	...	...	...	...	...
	Driving axles (carbon steel),	80,000	40,000	18	...	...
	,, (nickel ,,),	80,000	50,000	25	45	...
,, forgings (a),	Soft or low carbon steel,	58,000	29,000	28	35	180°, D = T.
	Carbon steel, not annealed,	75,000	37,500	18	30	180°, D = 3 T.
,, tyres,	Passenger engines,	100,000	...	12	...	...
	Freight engines and car wheels,	110,000	...	10	...	...
	Switching engines,	120,000	...	8	...	...
Structural steel for buildings,	Rivet steel,	50,000 to 60,000	30,000	26	...	180°, flat.
	Medium steel,	60,000 ,, 70,000	35,000	22 (d)	...	180°, D = T.
,, ,, bridges and ships,	Rivet steel,	50,000 ,, 60,000	30,000	26	...	180°, flat.
	Soft steel,	52,000 ,, 62,000	32,000	25	...	180°, ,,
	Medium steel,	60,000 ,, 70,000	35,000	22 (d)	...	180°, D = T.
Open hearth boiler plate and rivet steel,	Flange or boiler steel,	55,000 ,, 65,000	33,000	25	...	180°, flat.
	Fire-box steel,	52,000 ,, 62,000	32,000	26	...	180°, ,,
	Extra soft steel,	45,000 ,, 55,000	30,000	28 (d)	...	180°, ,, (e)
Steel rails (b),	. . . .	...	...	...	...	...
,, splice bars,	. . . .	54,000 to 64,000	32,000	25	...	180°, flat.

(a) In the four classes of steel forgings not tabulated above, it will be found, by reference to the specifications, that elastic limit is specified instead of yield point.

(b) This table does not include the drop test required on castings, axles, tyres, and rails; the homogeneity test for fire-box steel; nor the percussive test for large steel castings.

(c) The elongation given for castings, axles, forgings, and tyres are on 2-inch the others on 8-inch gauged length.

(d) Reference to the text of the specifications must also be made for the variation in elongation allowed for thin and thick structural steels and boiler steel, and for the tensile tests of full-sized eye-bars.

(e) These same bending tests are required after quenching. (See Specification for details.)

TABLE XCVIIIa.—Steel Forgings (American Standard).

Process of Manufacture.

1. Steel for forgings may be made by the open hearth, crucible, or Bessemer process.

Chemical Properties.

2. There will be four classes of steel forgings, which shall conform to the following limits in chemical composition :—

	Forgings of Soft or Low Carbon Steel.	Forgings of Carbon Steel not Annealed.	Forgings of Carbon Steel, Oil Tempered or Annealed.	Forgings of Nickel Steel, Oil Tempered or Annealed.
	Per cent.	Per cent.	Per cent.	Per cent.
Phosphorus shall not exceed	0·10	0·06	0·04	0·04
Sulphur ,, ,,	0·10	0·06	0·04	0·04
Nickel ,, ,,	...	...	...	3·75

Physical Properties.

3. The minimum physical qualities required of the different sized forgings of each class shall be as follows :—

Tensile Tests.				
Tensile Strength.	Yield Point.	Elongation in 2 Inches.	Contraction of Area.	
Lbs. per Sq. Inch.	Lbs. per Sq. Inch.	Per cent.	Per cent.	*Soft Steel or Low Carbon Steel.*
58,000	29,000	28	35	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
				Carbon Steel, not Annealed.
75,000	37,500	18	30	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
	Elastic Limit.			*Carbon Steel, Annealed.*
80,000	40,000	22	35	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
75,000	37,500	23	35	For solid forgings, no diameter to exceed 20 inches, or thickness of section 15 inches.
70,000	35,000	24	30	For solid forgings, over 20 inches diameter.
	Yield Point.			*Carbon Steel, Oil Tempered.*
90,000	55,000	20	45	For solid or hollow forgings, no diameter or thickness of section to exceed 3 inches.
85,000	50,000	22	45	For solid forgings of rectangular sections not exceeding 6 inches in thickness, or hollow forgings, the walls of which do not exceed 6 inches in thickness.

TABLE XCVIII*a*.—Steel Forgings (*Continued*).

Tensile Tests.				
Tensile Strength	Yield Point.	Elongation in 2 Inches.	Contraction of Area.	
Lbs. per Sq. Inch.	Lbs. per Sq. Inch.	Per cent.	Per cent.	*Carbon Steel, Oil Tempered—Continued.*
80,000	45,000	23	40	For solid forgings of rectangular sections not exceeding 10 inches in thickness, or hollow forgings, the walls of which do not exceed 10 inches in thickness.
				Nickel Steel, Annealed.
80,000	50,000	25	45	For solid or hollow forgings, no diameter or thickness of section to exceed 10 inches.
80,000	45,000	25	45	For solid forgings, no diameter to exceed 20 inches, or thickness of section 15 inches.
80,000	45,000	24	40	For solid forgings, over 20 inches diameter.
				Nickel Steel, Oil Tempered.
95,000	65,000	21	50	For solid or hollow forgings, no diameter or thickness of section to exceed 3 inches.
90,000	60,000	22	50	For solid forgings of rectangular sections not exceeding 6 inches in thickness, or hollow forgings, the walls of which do not exceed 6 inches in thickness.
85,000	55,000	24	45	For solid forgings of rectangular sections not exceeding 10 inches in thickness, or hollow forgings, the walls of which do not exceed 10 inches in thickness.

4. A specimen, 1 inch by ½ inch (1″ × ½″), shall bend cold at 180° without fracture on outside of bent portion, as follows :—

Bending Test.	
Around a diameter of ½ inch,	For forgings of soft steel.
½ ,,	,, carbon steel, not annealed.
1½ ,,	,, ,, annealed, if 20 inches in diameter or over.
1 ,,	,, ,, ,, if under 20 inches diameter.
1 ,,	,, ,, oil tempered.
½ ,,	,, nickel steel, annealed.
1 ,,	,, ,, oil tempered.

Solid Matter in Solution in Water.—Professor Vivian Lewes gave the following analyses as fairly representing the salts, etc., deposited from (1) ordinary river water, (2) in tidal rivers near their estuaries and well water near the sea, and (3) in ordinary sea water :—

TABLE XCIX.

Constituents.	River.	Brackish.	Sea.
Calcic carbonate,	75·85	43·65	0·97
,, sulphate,	3·68	34·78	85·53
Magnesic hydrate,	2·56	4·34	3·39
Sodic chloride,	0·45	0·56	2·79
Silica,	7·66	7·52	1·10
Oxides of iron and alumina,	2·96	3·44	0·32
Organic matter,	3·64	1·55	trace
Moisture,	3·20	4·16	5·90
	100·00	100·00	100·00

The following were given by him as analyses of the deposits found in a marine boiler using salt water as its extra supply and the engine getting its internal lubrication in the usual way with *valvoline*, which is a mineral oil having a specific gravity of 0·889 and the boiling point 700° Fah. It is quite free from free acids, and should be quite free from any animal or vegetable oil or fat, and therefore highly suitable for such a purpose.

TABLE C.—Boiler Deposits.

Scale from Furnace.			Deposit from Tubes.		
	From Top.	From Below.		Scale on Tubes.	Deposit above Scale.
Calcic sulphate,	84·87	59·11	Calcic sulphate,	50·92	11·60
,, carbonate,	5·90	6·07	,, carbonate,	4·18	0·82
Magnesic hydrate,	2·83	11·29	Magnesic hydrate,	14·12	22·21
Iron, alumina, and silica,	2·37	2·85	Iron, alumina, silica, &c.,	7·47	9·14
Organic matter and oil,	3·23	19·54	Organic matter and oil,	21·06	50·20
Moisture,	0·80	1·14	Moisture,	1·17	4·23
Alkalies,	nil	nil	Alkalies,	1·08	1·80
	100·00	100·00		100·00	100·00

Deposit from Bottom of Boiler.

Calcic sulphate,	22·52	Moisture,	5·79
,, carbonate,	nil	Alkalies,	1·80
Magnesic hydrate,	7·09		
Silica, alumina, and iron,	34·85		
Organic matter and oil,	27·95		100·00

On careful examination of the organic matter and oil present in these deposits, it was found that quite one-half of it was "valvoline," in an unchanged condition, which had collected round small particles of calcic sulphate.

The Effect of Temperature on the Strength of Metals.—To-day, with high pressure and high superheating likely to become general, it is of very great importance to know how far the metals used by engineers are affected by rise of temperature both in strength and structure. Steam (saturated) at a pressure of 150 lbs. above the atmosphere has a temperature of 365° Fah.; at a pressure of 200 lbs. the temperature is 387° Fah.; at 250 lbs., 406° Fah., and at 300 lbs., 421° Fah. Superheating may raise the temperature almost to any degree, and on shore as much as 600° Fah. is considered to be desirable. In any case, however, *within* the marine H.P. cylinder the temperature will not greatly exceed that of the saturated steam of the same pressure as in the valve-box. It may, therefore, be taken as a rule for such, that the H.P. cylinder and its internal fittings must be suitable for a temperature of 500° Fah.; the medium-pressure cylinder for 350° Fah., and the low-pressure cylinder for 250° Fah. and the condenser top, 200° Fah.

Cast Iron is little affected by such temperatures as the above. Tredgold stated that cast iron was by no means reduced in strength by heat up to 600° Fah.

Wrought Iron and Steel (Mild Ordinary) gain in ultimate strength with rise of temperature, so that at 400° Fah. it is 10 to 15 per cent. higher than when at 60° Fah., but with a reduction in the extension.

Copper.—Pure copper is seldom or never used in modern engineering. Commercially, pure copper, as in pipes and sheets, is not much affected by heat up to 500° Fah. Sir W. Roberts-Austen found that very pure copper having a tensile strength of 29,600 lbs., with 27 per cent. extension at 59° Fah., had 30,900 lbs. with 30 per cent. at 212°; 30,090 lbs. with 23 per cent. at 304°; 29,020 lbs. with 25 per cent. at 424°; 28,240 lbs. with 37 per cent. at 480°; and 25,390 lbs. with 21 per cent. at 617°. Also that copper containing only 0·2 per cent. of arsenic had an ultimate strength of 30,150 lbs. with 30 per cent. extension at 64° Fah., and 30,880 lbs. with 30 per cent. at 214°; 30,960 lbs. with 21 per cent. at 290°, and 28,730 lbs. with 30 per cent. at 480°. Such copper as this is very suitable for steam pipes. Copper containing 0·5 per cent. of arsenic is also good for locomotive fire-box plates as at 835° Fah., as its tensile strength is as much as 20,000 lbs., with 13 per cent. extension. Professor Unwin states that rolled copper, which, at 60° Fah., had a strength of 40,000 lbs., had 38,500 at 212°, 36,000 at 350°, and 33,000 at 536°.

Gunmetal or Admiralty Bronze,* which at 60° may be taken as having a tensile strength of 15·3 tons with an extension of 12·5 per cent., has 31,000 lbs. with 10 per cent. at 200°; 29,800 lbs. with 8·25 at 350°; but only 15,700 lbs. with 0·75 per cent. at 400°; at 500° Fah. the strength is about the same, but with no extension whatever. Bronze with *more zinc* in its composition, say, 15 per cent., has a tensile strength of 29,000 lbs. with an extension of 26 per cent.; at 400° Fah. it is still 27,750 lbs. with 25 per cent., but at 450° it drops to 9,700 lbs. with 1·2 per cent. only.

Naval Brass, which at 60° has a strength of 60,000 lbs. with an extension of 19 per cent., still has 52,000 lbs. with 16 per cent. at 400°; at 500° the strength is about the same, with an extension of over 13 per cent.

* Mr. J. Dewrance found that an alloy of 87·5 copper, 10 tin, 2 zinc, and 0·5 lead had a tensile strength of 15·8 tons with an elongation of 18 per cent. at 550° F., and at 700° 8·25 tons with 2 per cent. When cold it had 16·5 tons with 8 per cent.

TABLE CI.—Effect of Heat on Metals.

Metal or Alloy.	At 100° F.		At 250° F.		At 350° F.		At 400° F.		At 450° F.		Melting Point. F.
	Ultimate Strength	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	Ultimate Strength.	Extension.	
		Per cent.		Per cent.		Per cent.		Per cent.		Per cent.	
Copper (0·2 per cent. arsenic),	30,325	30·3	31,000	26·2	30,950	21·1	29,750	27·0	29,000	29·0	2,000°
Admiralty bronze,	32,000	12·0	30,800	9·3	29,800	8·25	15,700	0·75	15,500	0·2	1,700°
Bronze (87·5 Cu, 10 Sn, 2 Zn, 0·5 lead),	36,900	8·0	36,700	8·0	36,400	10·0	36,300	11·0	36,000	12·2	1,700°
Mercantile bronze (soft), cast,	29,000	26·0	28,800	25·8	28,000	23·0	27,750	25·0	9,700	1·2	1,680°
Naval brass, rolled,	58,700	18·8	54,500	17·6	52,600	16·6	52,000	16·0	52,000	14·0	1,640°
Phosphor bronze, **W** brand, cast,	39,000	17·8	37,000	15·0	29,400	7·0	27,000	5·0	27,000	5·0	1,900°
Phosphor bronze, **W** brand, rolled,	68,250	6·8	68,000	7·0	67,750	7·5	64,500	8·1	60,000	9·0	1,900°
Parsons' M. Bronze Co., immadium, cast,	80,600	15·0	..	..	..	..	56,400	25·0	51,500	22·5	1,660°
Aich's metal, rolled,	55,250	..	47,000	..	12,300	..	39,750	..	37,000	..	1,650°
Bull's metal, rolled,	70,000	16·0	63,600	13·6	58,700	13·0	56,750	14·6	..	..	1,800°
Aluminium, rolled,	25,300	..	20,250	..	15,500	..	13,300	..	11,500	..	1,160°
Melloid, rolled,	98,000	12·0	..	..	..	..	90,700	15·0	87,300	15·0	..

Note.—Copper at 500° has a tensile strength of 10·8 tons, and extension 40 per cent., while at 650° it is 9·7 tons with 33·5 per cent. in 3 inches.

The bronze with 0·5 per cent. of lead has at 550° F. 15·8 tons and 18 per cent.

Phosphor Bronze (cast) with a tensile strength of 39,000 lbs. and an extension of 17·5 per cent. at 60° Fah. has about the same strength at 200°; 37,000 lbs. with 15 per cent. extension at 250°; 36,700 with 12 per cent. at 300°; 29,400 with 7 per cent. at 350°; and 27,000 with 5 per cent. at 400° Fah., and the same at 500° Fah. Rolled phosphor bronze, with a strength of 68,500 lbs., and an extension of 6·3 per cent. and an *elastic limit* of 62,000 lbs. has 67,500 lbs. with 8·2 per cent., and an elastic limit of 64,500 lbs. at 400° Fah.; and at 500° Fah. the strength is as high as 60,000 lbs. with 9·0 per cent. extension, the elastic limit falling to 54,000 lbs. only.

Bull's Metal, of soft quality, rolled, has a strength of 73,000 lbs. with an extension of 16·4 per cent., the elastic limit being 64,000 lbs.; at 300° Fah. it stood 60,700 lbs. with 12·7 per cent. extension, and at 400°, 56,750 lbs. with 14·6 per cent., the elastic limit being then as high as 47,000 lbs.

Aich's Metal, which at 60° Fah. has a tensile strength of 57,300 lbs. and 49,600 lbs. at 212° Fah.; it has still 35,500 lbs. at 482° Fah.

Aluminium, which at 60° Fah. has a tensile strength of 26,000 lbs., has at 212° Fah. 21,280 lbs., at 300° Fah. 18,100 lbs., at 390° Fah. 14,100 lbs., and at 480° 10,750 lbs.

Manganese Bronze.—Parsons' No. 6 metal, which has a tensile strength of over 76,000 lbs., with an ultimate extension of 25 per cent. at 100° Fah., still has 56,000 lbs. at 450° Fah., with an extension of 27 per cent., and is specially designed for boiler and cylinder mountings and other castings exposed to high temperatures.

Melloid, a special alloy made by Bull's Metal Company to resist heat, has an ultimate tensile strength at 60° Fah. of 98,000, with an extension of 12 per cent. in 2 inches; at 600° Fah. it is 79,000 lbs., with 18 per cent.; at 700° Fah. it is 51,900 lbs., with 22 per cent.; and even at 800° Fah. it could resist 31,800 lbs. and extend 50 per cent.

Dewrance's Bronze, an alloy of 87·5 copper, 10 tin, 2 zinc, and 0·5 lead, as cast, has a tensile strength of 16·5 tons, or 36,900 lbs. with an extension of 8 per cent. At 450° Fah. the tensile is 36,060 lbs. with an extension of 12·2 per cent., while at 550° it is as high as 35,400 with 18 per cent.; and even at 600 the tensile was 25,200 lbs. with 8 per cent.

Phosphor Bronze, 96·1 copper, 3·6 tin, and 0·3 phosphorus, is good for temperatures up to 450° Fah., having an ultimate tensile of 25 tons, with elongation of 15 per cent. in 6 inches, and a yield point of 21 tons. After exposure for 300 hours to this temperature the ultimate was still 23 to 24 tons with 15 per cent.

Manganese Copper, 96·5 copper, 3·5 manganese, has a yield point of about 18 tons, and is good for temperatures up to 600° Fah.; the elongation at ultimate is 17 per cent. in 2 inches.

Nickel Copper is also a useful alloy for things exposed to high temperatures, and may be used for castings or drawn bars.

CHAPTER XXX.

OIL AND LUBRICANTS—ENGINE FRICTION.

In these days of large engines and high-speed reciprocators and still higher turbines, the quality of the lubricant used is of first importance. Notwithstanding this, it is by no means an uncommon thing to find that a shipowner, after paying a considerable price to have a first-class engine, will supply the engineer with an inferior oil for the sake of saving 2d. a gallon. The falseness of this economy is apparent when it is known that there may be a difference of as much as 5 per cent. in the efficiency of an engine by using an inferior oil as against a good one, and that with inferior oil the wear and tear of brasses, etc., may be also serious, whereas, with a proper supply of a good lubricant there should be really no wear or tear. At the same time, it must be admitted that the method of lubrication hitherto generally observed on board ship is a primitive and likewise very wasteful one. Surely some system of continuous circulation of the same oil might be followed with the same good results and certainty as obtain with high-speed engines on shore. But it is quite true, however, that to attain this end the use of water on the bearings must first be stopped entirely as a general practice, and only resorted to on emergency; no doubt this course can be safely followed if the oil is passed through the bearings, pins, etc., as lavishly as is the case with the electrical engine. In the meanwhile it is quite possible and comparatively easy to devise a system whereby the oil from the bearings can be collected, separated from the dirt and water, and so dealt with as to be used over again; it could in that case be supplied much more freely than is usual now with the lubricating gear at present fitted to modern machinery.

The Consumption of Oil for the lubricating of the main engines of a steamship should with reasonable care not exceed 1 gallon per 100 I.H.P. per 24 hours' steaming. But if the engines are fitted with a good and efficient system of forced lubrication, including coolers and filters, and charged with a supply of oil equal to two or three days' consumption at the above rate, they may be run for months with little or no additional lubricant with the smallest amount of attention from the engine-room staff. The saving of oil alone amply repays interest on the additional cost.

Oils are of three kinds—animal, vegetable, and mineral—and have probably been discovered and used by mankind in this order. The two former are called fixed oils, because they will not volatilise without decomposition, as do the mineral.

Of Animal Oils, the ones most used by engineers are—neat's foot, lard, whale, and sperm.

Sperm Oil is obtained from the head of the sperm whale, and is a very high-class lubricant, in fact, there is none better for guides and bearings, etc., of engines of all sizes. It is, however, of high price, and there are other mixed oils almost as good for marine purposes to be obtained at a much lower price.

Whale Oil, better known in other days as train oil, is not often used by itself as a lubricant, although, of course, it could be on an emergency if properly refined so as to be free from any substance that would easily solidify and become hard like a gum. It is now generally used for tempering large steel forgings, etc.

Lard Oil has not been used on board ship in this country as a lubricant, but in America, as also nowadays in this country, it is known as a good lubricant, having a viscosity equal to that of olive oil, and found to be the best oil for using on the quick cutting modern machine tools when being forced to their utmost power.

Neat's Foot Oil, obtained from the fat of the feet and hocks of bullocks, was a favourite one with older engineers, and is still used by oil manufacturers to mix with other oils, usually mineral, so as to form a lubricant suitable for general purposes.

Vegetable Oils.—The ones best known to engineers are—Olive, castor, colza, rape, palm, cotton-seed, linseed.

Linseed Oil, obtained from the seed of flax, is not suitable for a lubricant, but is very useful to the engineer in other ways.

Cotton Oil is expressed from the seed of the cotton plant; when refined it is colourless and tasteless. It may be used instead of lard oil for cutting tools, as also for mixing with other oils. It is rather too volatile for ordinary marine purposes, and, in fact, from the tendency to dry up, it is hardly looked upon as a safe lubricating oil.

Palm Oil, in the form of grease, was formerly sometimes used for the outer bearings of paddle ships, and for the filling of the centre recess of tunnel shafting bearing blocks. It may still be used for these purposes instead of tallow on stations where it is cheap. It is, however, now hardly in the purview of the marine engineer as an every-day lubricant.

Rape and Colza Oils, expressed from the seed of rape or wild turnip, are very much liked, and used generally by locomotive engineers; apparently generally for the reason that, while being good lubricants, they also are good lamp oils, and not liable to the dangers of fire as is petroleum when an accident occurs, a matter of great consideration before the days of train lighting by gas and electricity.

Castor Oil, extracted from the castor oil bean, is largely used on board ship in the East Indies, where it is cheap as well as plentiful. It is a very good lubricant, and may be used with advantage, especially in hot climates, where it, as other oils, is apt to run thin. It may here be mentioned that the lubricating value of an oil depends very much on the temperature at which it works.

Olive Oil, expressed from the fruit of the olive tree, has been used as a lubricant on board ship very extensively, and is a most satisfactory one, being suitable to bearings and guides of all sizes. At one time it was the only oil used in the British Navy, and when replaced by Rangoon oil by an economy-seeking Board, it was shown to be not so extravagant as the price seemed to indicate. Olive oil solidifies at 36° Fah., and, like castor, has good viscosity at fairly high temperatures.

Mineral Oils.—These oils, quite unknown to the old engineers, are now in very general use both for internal and external lubrication, and seem destined to oust other oils from the marine engine-room. The chief supplies are from Southern Russia (Baku) and from the United States of America, although petroleum is known to exist in almost every part of the world. In this country we have a somewhat similar article in our Scotch shale oils, obtained by the distillation of shale and separation from the paraffin contained therein.

The petroleum, as obtained from the wells, is a thick, treacly substance, Russian having a specific gravity as high as 0·938, and the American 0·886. After refining, the residue known in Russia as *Astatki*, is used for fuel on board ship and on locomotives. The American residuum has a specific gravity of 0·928, and is also used for fuel.

Between that and the highly inflammable light oils, such as petrol, benzaline, etc., are a large number of grades having varying densities as well as viscosities, and distilled at different temperatures. Among them are oils suitable for the heavy bearings of marine engines as well as the light ones of the smaller kinds; also oils whose evaporating point is high, and, therefore, suitable for the internal lubrication of marine engines. All these refined oils are pure hydrocarbons—that is to say, they are composed only of hydrogen and carbon. They will not saponify in the ordinary way nor form fatty acids at all, and, therefore, may be used safely for internal lubrication without fear of damage to the castings, or to condenser and boilers. At the same time, it is, of course, advisable to prevent the passage of any oily matter whatsoever to the boiler, whether it be some of the pure oil carried along without destruction, or the refuse after use in the engine.

The following tables give the leading characteristics of the above oils:—

TABLE CII.—Specific Gravity of Oils when at 60° F.

Archbutt and Deeley.

Sperm oil,	0·880
Neat's foot oil,	0·915
Lard oil,	0·916
Whale oil,	0·922
Seal oil,	0·925
Porpoise oil,	0·926
Olive oil,	0·915
Rape-seed or Colza oil,	0·915
Cotton-seed oil,	0·923
Castor oil,	0·965
Palm oil,	0·923
Rosin oil,	0·980
Scotch lubricating oils,	0·875 to 0·895
American dark machinery oils,	0·877 „ 0·887
„ light „	0·897 „ 0·920
„ cylinder „	0·885 „ 0·905
Russian pale for light machinery oils,	0·895 „ 0·903
„ heavy „	0·903 „ 0·910

TABLE CIII.—Density of Oils at Various Temperatures.
Archbutt and Deeley.

Name of Oil.	60° F.	100° F.	150° F.	212° F.
Sperm oil,	0·8783	0·8637	0·8455	0·8229
Olive oil,	0·9159	0·9011	0·8826	0·8596
Rape or Colza oil,	0·9151	0 9005	0·8822	0·8595
Castor oil,	...	0·9473	0·9284	0 9050
Scotch mineral oil, "865,"	0·8683	0·8533	0·8347	...
,, "890,"	0·8905	0·8761	0·8581	...
Russian mineral light machinery oil,	0·8978	0·8837	0·8661	0·8442
,, heavy ,,	0·9096	0·8957	0·8784	0·8568
American mineral spindle oil,	0·8677	0·8535	0·8358	0·8138
,, red engine oil,	0·9162	0·9020	0·8843	0·8624
,, Bayonne engine oil,	0·9113	0·8973	0·8797	0·8579
,, dark medium machinery oil,	0·8839	0·8695	0·8514	0·8291
"Valvoline,"	...	0 8757	0·8587	0 8377
Dark filtered cylinder oil, "900,"	...	0·8883	...	0·8490

The Flash Point of American cylinder oil (sp. gr., 0·902) is 585° F.
,, ,, ,, (,, 0·893) is 424° F.
,, ,, machinery oil (,, 0·897) is 402° F.
,, Russian ,, (,, 0·909) is 380° F.

The Boiling Point of mineral oils is unusually high, and in the case of refined distilled oils seldom less than 600° F., that of linseed oil being 602° F. The American oils are seldom fluid at a temperature under 25° F., while the Russian generally remain so as low as 0° F.

Rape oil solidifies at freezing point (32° F.), lard oil at 40° F., sperm oil at 39° F., and olive oil at 36° F.

TABLE CIV.—Viscosity of Oils at Various Temperatures (Water at 68° F. being 0·01028)—*Archbutt and Deeley.*

Name of Oil.	60° F.	100° F.	150° F.	212° F.
Sperm oil,	0·420	0·185	0·085	0·046
Olive ,,	1·008	0·377	0·154	0·070
Rape or Colza oil,	1·118	0·422	0·177	0·080
Castor oil,	...	2·729	0·605	0·169
Scotch mineral oil "865,"	0·146	0·066	0·036	...
,, ,, "890,"	0·509	0·183	0·069	...
Russian light machinery oil,	1·156	0·307	0·099	0·043
,, heavy ,,	3·592	0·762	0·196	0·066
American spindle oil,	0·727	0·236	0·086	0·039
,, pale ,, "903/7,"	1·138	0·342	0·115	0·049
,, ,, ,, "907/12,"	1·479	0·419	...	...
,, red engine oil "910/20,"	1·915	0·496	0·150	0·058
,, "Bayonne" engine oil "912/15,"	2·172	0·572	0·173	0·063
,, dark medium machinery oil,	3·046	0·705	0·210	0·076
"Valvoline,"	...	2·406	0·605	0·187
American filtered cylinder oil "888,"	...	3·702	...	0·238
,, dark ,, ,, "900,"	...	6·264	...	0·314

TABLE CV.—Boiling, Setting, and Flash Points.

Name or Brand of Oil.	Sets at	Flashes at	Boils at
	Degrees F.	Degrees F.	Degrees F.
American cylinder, sp. gr. ·902,	25	585	650
,, ,, ,, ·893,	25	424	600
Valvoline ,,	...	...	700
American machine, sp. gr ·897,	25	402	600
Russian ,, ,, ·909,	0 to 10	380	...
Scotch shale oil,	32	...	...
Cotton-seed oil,	34 to 50	...	580
Castor oil,	14	...	556
Lard oil,	25 to 42	...	..
Linseed oil,	5 ,, 17	...	574
Neat's foot oil,	32 ,, 50	...	...
Olive oil,	21 ,, 39	..	...
Rape or Colza oil,	10 ,, 28	...	625
Sperm oil,	32	450	600
Palm oil,	81 to 106	...	...
Whale oil,	32	440	620
Vaseline,	104 to 120	640	...
Tallow (Russian),	107 ,, 119	540	660

TABLE CVI.—Price per Gallon of Various Oils.*

Name of Oil.	Price per Gallon.	Name of Oil.	Price per Gallon.
Castor oil,	2/3 to 2/6	Palm oil,	2/9 to 3/-
Colza oil,	2/1	Sperm oil (common),	2/3 ,, 2/6
Rape oil,	2/-	Whale oil,	2/-
Cotton-seed oil,	2/-	Neat's foot oil,	3/- to 3/3
Linseed oil,	2/- to 2/3	Lard oil,	4/- ,, 4/3
Olive oil,	3/3 ,, 3/9	American lubricating (good),	1/3 ,, 1/6

Compound Oils.—The oils most generally used on shipboard for external lubrication are compounded of a mineral oil and one or more of the fixed oils. Mineral oil of itself, although, on the whole, a good lubricant, is not suitable for bearings or guides which have, or may have, a greater pressure per square inch than 70 lbs. Moreover, increasing the viscosity of a mineral oil by other denser oils does not improve it for the purpose, and, in fact, is often the means of causing greater friction. If, however, a fixed or fatty oil be compounded with the mineral oil in a proper way so that they do not separate out when standing still, the compound will form a suitable lubricant for higher pressures than the above. The higher the pressure per square inch, the greater viscosity must be given to the lubricant by adding fatty oils of higher viscosity. It will also be observed by reference to Table civ. that mineral oils whose viscosity is very high at 60° Fah. are the very ones whose viscosity rapidly decreases with an increase of temperature, and although the fatty oils decrease in viscosity as the temperature rises, their

* These are all pre-war prices, now they are abnormal and variable.

rate of fall is not so great. In other cases, however, pure mineral seems to be lacking in what Messrs. Archbutt and Deeley term "oiliness."

Mineral oils are sometimes thickened in viscosity by dissolving certain soaps in them, the favourite for this purpose being aluminium soap, which is made by saponifying whale, cotton, or even lard, oils, and caustic soda, and pouring into it when hot a solution of common alum. This is, of course, an adulteration, but not altogether a bad one, inasmuch as the oil then sticks to the journal much better than it did in the pure state, and gives a better lubrication to the bearing. The soap, however, is liable to start out and obstruct the oilways. When mineral oil has been treated in this way it can generally be detected by the tendency to threadiness when the cork is drawn from the bottle of the mixture.

The oils most generally used for compounding with mineral oils are—Rape, olive, lard, neat's foot, whale, and sperm. Castor oil will not dissolve in and compound with a mineral oil unless it is first treated with an equal volume of tallow oil. It is, however, not often used for the purpose. Rosin oil is sometimes used to increase the specific gravity of a compound oil as well as to improve its viscosity. It does not, however, improve its lubricating powers, and as it increases the tendency to gumminess it should be avoided. Marine engineers usually prefer the compound oil to a pure mineral one on account of the tendency to lather—that is, to form grease by mixing with the water applied to the bearing. This formation is hastened, of course, by the use of a little soda in the water, and when formed it generally prevents the oil running freely out of the bearing or guide. Besides which, should the bearing or guide become warm, this grease melts and adds to the quantity of lubricant so as often to check any further heating. It is evident, therefore, that for a system of forced lubrication the oil must not be able to saponify or even to form an emulsion easily. Certain kinds only of mineral oil can be depended on for the purpose, as being free from these vices.

Soaps.—Pure mineral oils are unaffected by alkalies and have little or no effect on metals. They, therefore, do not saponify, and for this reason are used for internal lubrication. Formerly, when tallow and fatty oils were used indiscriminately for internal and external lubrication, the condenser very soon got covered with a soapy deposit more or less hard. All vegetable and animal oils contain fatty matters, which combine with alkalies and form what are called soaps. Some oils will also combine with certain salts of metals and form a soapy mixture; for example, white lead and linseed oil. Oils combined with caustic soda make hard soaps, and, when combined with caustic potash, form what is known as soft soap. Solutions of these soaps will act as lubricants, but are, of course, not so efficient as the oils. They are sometimes temporarily used to cleanse oil ways of the gummy greasy matter which may have been deposited from the oil.

Adulteration of Oils.—Compound oils are generally valued by their density and viscosity. It is, therefore, the aim of the dishonest merchant to add some cheap ingredient which shall effect this object. That most generally used for the purpose is resin oil, being cheap and easily mixed with other oils. It is, however, itself inferior as a lubricant, and has the fatal vice of drying hard like varnish. Its presence in oils is easily detected by the pungent taste, which is unmistakable,* and its failure to saponify when an alkali is added.

Solid Lubricants.—There are certain substances which act as lubricants, although not at all on the same principle as does a liquid. The chief of these are powdered soapstone, flour of sulphur, and plumbago.

Soapstone and French chalk, finely powdered to a flour, has a greasy feel, and acts as a very efficient lubricant to wood fittings, and also, to some extent, to metallic ones. It is, however, of little practical value as a modern engine lubricant.

Flour of Sulphur was frequently used by the older engineers, whose acquaintance with hot bearings was much more intimate and frequent than is that of their succ ssors to-day; they always carried it among their stores, and, when a bearing was troublesome, poured a small quantity down the oil holes. Its efficiency probably varied with the heat of the bearing. Its function seems to have been to combine with the brass dust and brass ridges, and thereby to form a sulphide or sulphides of the metals which was much softer than the metals themselves, and, in a plastic state, permitted of the production of a web which lined the bearing and provided a new surface for the journals to run on. If this theory is a correct one, the material, of course, cannot be considered as a true lubricant.

Graphite or Plumbago.—This material, when pure and so highly subdivided that it forms an impalpable powder, has an extremely greasy feel to the fingers, and is an exceedingly good lubricant for both metals and woodwork. It may be used together with oil, grease, or vaseline for large and heavy bearings revolving at moderate velocity. It is also used for the internal lubrication of some engines, and doubtless may, in the near future, play an important part in that direction when superheated steam is used on shipboard. At present the means of applying it to the internal parts are crude and unsatisfactory. Ingenuity, however, will soon find a method to overcome the present difficulties, and, whether this material is carried over in a dry state or mixed with vaseline or some other semi-liquid carrier, it will be properly distributed and do good service. A small quantity of graphite may be used with advantage on bearings which are lubricated with oil, as it tends to keep the surface of both journal and bearing smooth and polished. When a brass has become rough or scored, this material will fill the recesses and present a new and good surface to the journal.

Tallow and certain other fats, both vegetable and animal, are, at ordinary temperatures in temperate zones, in the solid state, but a slight rise of temperature converts them into a liquid, and, therefore, it is idle to deal with them as if they were solid lubricants. Moreover, none of them are of great importance to the marine engineer of to-day.

Grease.—If oils whose specific gravity is nearly that of water be mixed with pure warm water and agitated, an *emulsion* is formed, and the oil will separate from the water very slowly indeed, even when the mixture is allowed to stand. If the water is made slightly alkaline and carefully stirred into some animal and vegetable oils, especially if at a moderately high temperature, they combine and form grease—the value of which as a lubricant depends in great measure on the quality and quantity of the oil used in

* For further particulars of oils, see *Lubrication and Lubricants*, by Archbutt and Deeley, published by C. Griffin & Co., Ltd.

making it. Tallow, palm oil, and other such fatty substances will take up very considerable quantities of water.

Rule.—Let S be the velocity of sliding in feet per minute, P the maximum steady working pressure in lbs. per square inch on a slide lubricated in the common way (that is, without force).

Then $$P = 50{,}000 \div S + 100$$

if intermittent as in piston-rod guide, maximum value is P, then

$$P = 65{,}000 \div S + 100.$$

For journals with steady load

$$P = 65{,}000 \div d \times R + 30.$$

d being diameter in feet, and R the revolutions per minute.

For intermittent circular as in crank-pins

$$P = 100{,}000 \div d \times R + 30.$$

For intermittent oscillating as in cross-head brasses

$$P = 250{,}000 \div R + 130.$$

TABLE CVII.—Coefficients of Friction.

Nature of the Surfaces.		Quite Dry.	Greasy Dry.	Well Lubricated with			Constant Flow of Oil.
				Oil	Tallow or Soap.	Water.	
Wood on wood (dry),		0·25 to 0·5	...	...	0·2 to 0·04	...	...
Metal on hard wood,		...	0·15	0·07 to 0·08	...	...	0·05
Metals on oak,	(Rk)	0·5 to 0·6	0·24 to 0·26	...	0·2	...	...
,, metals only,	,,	...	0·15 to 0·2	...	0·14	0·3	...
,, ,, smooth,	,,	...	0·07 to 0·08	0·05	...	...	0·03
Bronze on lignum vitæ,	,,	...	...	...	...	0·05	...
Iron ,, ,,		...	...	...	...	0·05	...
Bronze on bronze sliding,	(S)	...	...	...	...	0·38	...
,, ,, ,,	(Rn)	...	0·175	...	...	...	...
Steel ,, ,,	,,	...	0·139	...	...	...	...
Wrought iron on bronze sliding,	,,	...	0·135	...	...	...	...
Cast ,, ,, ,,	,,	...	0·141	...	...	...	...
Steel on cast iron	,,	...	0·151	...	...	...	...

TABLE CVII.—Coefficients of Friction—*Continued.*

Nature of the Surfaces.	Quite Dry.	Greasy Dry.	Well Lubricated with Oil.	Well Lubricated with Tallow or Soap.	Well Lubricated with Water.	Constant Flow of Oil.
Steel on wrought iron, ... (Rn)	..	0·189	...	...	...	...
Wrought iron on cast iron, ... ,,	...	0·170	...	...	...	...
,, ,, tin, ... ,,	...	0·181	...	...	...	...
,, ,, magnolia, . . (RHS)	...	0·060	...	...	...	...
,, ,, white metal,. . .	...	...	0·025	...	...	0·01 to 0·015
Loco axles, white metal, ... (W)	...	...	0·017	...	...	...
Shafts on bronze, ... (Rn)	...	...	...	0·028	...	...
Leather on metals, ... (Rk)	0·56	0·23	0·15	...	0·36	...
Wrought-iron shaft in cast-iron bearing, speed 10 feet per minute, . . .	...	...	0·079	...	...	...
Wrought-iron shaft in cast-iron bearing, speed 60 feet per minute, . . .	...	...	0·052	...	...	...
Wrought-iron shaft in cast-iron bearing, speed 100 feet per minute, . . .	...	...	0·050	...	...	...
Load 50 lbs. per square inch, speed 50 feet, (G)	...	...	...	...	...	·0032
,, 50 ,, ,, ,, 110 ,, ,,	...	...	...	...	...	·0064
,, 50 ,, ,, ,, 190 ,, ,,	...	...	...	..	...	·0106
,, 150 ,, ,, ,, 50 ,, ,,	...	...	...	...	...	·0017
,, 150 ,, ,, ,, 110 ,, ,,	...	...	...	...	...	·0024
,, 150 ,, ,, ,, 190 ,, ,,	...	...	...	...	...	·0047
Speed of 10 feet per min., load 50 lbs., . ,,	...	...	...	...	...	·0009
,, 10 ,, ,, 75 ,, . ,,	...	...	...	...	...	·0007
,, 10 ,, ,, 150 ,, . ,,	...	...	...	...	...	·0250
,, 15 ,, ,, 50 ,, . ,,	...	...	...	...	...	·0012
,, 15 ,, ,, 150 ,, . ,,	...	...	...	...	...	·0051
,, 7·8 feet per minute, moderate load, ,,	...	0·094	0·072	...	..	·064
,, ,, ,, heavy ,, ,,	...	0·051	0·046	...	...	·047
Railway axles and bearings (static), .	...	...	0·100	0·067	...	...
Cast-iron steel tyres, ... (Gl)	0·36 to ·048	5 to 60 miles per hour.				
Steel tyres on steel rails, ... ,,	0·11 to 0·04	10 to 50 ,, ,,				

Note.—Rk stands for Rankine; Rn, for Rennie; S, for Summers; W, for Wood; G, for Goodman; RHS, for Prof. Smith; Gl, for Gallon and Westinghouse.

CHAPTER XXXI.

TESTS AND TRIALS: THEIR OBJECTS AND METHODS.

THAT there must be tests and trials of ships and machinery goes without saying, but their nature and extent is a matter of considerable controversy, and one which has given rise to much discussion. In the case of tests of material only, samples are, as a rule, dealt with, which can be and generally are subject to trials ending in destruction, during which process certain facts and figures are observed and noted, so that from them conclusions may be drawn as to the fitness of the material of which they formed portions for the purposes designed. Seldom or never is any actual portion of an engine or boiler subject to any mechanical test until fitted in place and under trial conditions, hence the stress on it is never greater than the working stress, except when the trial is by water pressure greater than that when working with steam.

It is a moot question as to which is the best criterion of the value of a material, the *ultimate tensile strength*, the *elastic limit*, or the *yield point*. The ultimate tensile may be taken as a very crucial test in demonstrating the elasticity, as well as the power of resistance; but it is after all the prelude to a destruction which is irresistible that is being observed, and, therefore, while being interesting, it bears no anology to what should take place in practice; in fact, it is not the barometer, but the storm and its effects. The elastic limit is really only a phase or milestone in the course of a test, and that the change in the relations between stress and strain is anything more than an interesting phenomenon remains to be proved, except that the stretching in becoming greater points to a giving way of tenacity for the time, for if the load is relieved and for a time withdrawn, the material can better resist the load when it is again applied. It is pretty certain that the yield point or that stress at which the material ceases to regain its original length when relieved of it is open to some objection as a criterion of its ability; and that it is more than probable that all metals after manufacture will have permanent set after the application of load far below those which afterwards fail to produce that phenomenon, just as a rope will stretch under quite light loads when quite new, and will continue to stretch until its component parts have been forced into the position which enables them to be efficient, and whereby the fibres can all take their proper share of resistance. In metallic structures a somewhat similar process goes on, but to a less notable extent, and some parts of each component is probably stressed beyond the yield point of the material; but gradually such stressing decreases until, like a built-up gun made scientifically, each lamen takes its fair share of the load which comes on the component when under working conditions.

On the whole, it would seem, from the practical engineer's point of view, that the yield point is the safest criterion and guide, especially if it is ascertained in ways analogous to service conditions. If an engineer desires to employ a particular metal, which will, when in use, be subject to certain maximum stress, then it should be tested with a load which produces double that stress, and remain applied for a considerable time, say 10 hours, after which there should be no permanent set whatever on its removal. Before this crucial trial, however, there should be the application of the load and its removal after, say, five minutes to take out the "initial set." This would show its ability to take continuous load; but for intermittent and alternating stresses such a load should be applied and released suddenly, say a dozen times, and the effect of each carefully noted. These are only rough and ready means of proving the general ability of the material to endure loads which, while being quite moderate, are much beyond those coming on in practice.

It has been also a subject of controversy whether engineers shall work with *factors* of a safety or *margins* of safety; that is, shall the test stress which proves the strength of a structure be a multiple of the working stress, or shall it be a fixed excess of it? The Board of Trade, Lloyd's Register, and some other public institutions stick to the factor of safety—in fact, so long as the Board of Trade follow this method, these others cannot help but follow. The British and other Admiralties, and most of the foreign institutions governing shipping, adopt the margin or a modification of it. That is, past a certain working pressure the addition to it for testing boilers, high-pressure cylinders, etc., is a constant quantity. If the object of a test is to prove the ability of a structure to withstand the effect of the greatest load which can come on it, there seems no good reason why the test should be exactly double the normal maximum pressure, although under some circumstances even that test might be too small. If the test is one appreciably beyond the maximum normal with the generality of engines and boilers, it should prove all that is necessary when there is nothing unusual in the design and construction. If their design and construction is abnormal, is complicated, so as to be impossible of being carefully calculated, and, moreover, when at work, is liable, as oil and gas engines are, to heavy shock and unexpected conditions, the test should be far beyond, and may be justifiably more than double the supposed working pressure. It is obvious, therefore, that in all tests and trials there must be elasticity in the making and application of rules, and each case must stand by itself; the great thing being to bear in mind what is the object of the trial, and to make the means subservient to the true end.

Tests and Trials are made of the materials and the parts of an engine, and further of the engine itself when complete in a more or less haphazard way, and generally without clear ideas as to the object or nature of each of them, and consequently they are often not so satisfactory or conclusive as they should be. The object of a trial of any kind should be kept clearly in view all the time when making it, and the end rather than the means should have the first attention; the nature of the trial or test must be always subservient to the particular object desired, and the mixing up of policies strictly avoided. In every-day practice there are rough and ready ways by which the engineer can satisfy himself as to the adequacy of the means he adopts and the safety of the materials he employs on them, but more than that is necessary for success in modern marine engineering where high speed and huge output of power involve forces and stresses complicated and heavy, continuously applied throughout the whole 24 hours without intermission or stoppage of machinery, which must not fail or even be feared.

The tests applicable to such mechanism are not necessarily severe, but they must be very circumspect, and used with discretion. The endurance of a material is a quality highly to be desired in most parts, while in others it is secondary to high elastic limit.

In some parts of an engine necessity compels the minimum of material with the maximum of stress as a consequence; in such a case it is imperative that the metal employed should have a high elastic limit, and ample proof of it must be insisted on from the testing machine. If the stresses are to be suddenly applied the metal must be shown to be capable of withstanding them by "drop" tests and other similar means. If they are intermittent, or if alternating, then simple tensile or drop tests are not sufficient; there must be a series of bending backwards and forwards tests, or other similar tests to prove the endurance of the metal. On the other hand, for such parts as are of necessity large to give bearing surface, and on which the stresses are always low, it seems folly to waste time and money on any tests beyond those which are involved in machining, etc., in process of manufacture; the nature of steel can be gauged fairly correctly by observing how it cuts in the lathe and planing machine. Such things as are materially affected by heat or other treatment, while undergoing manufacture into the finished article, should have test pieces removed after rather than before such processes. But where such a process as oil tempering is carried out with the object and certainty of improving them, no such test is necessary.

In the case of cast iron some tests are necessary to prove the quality rather than the actual tensile strength; that is, the ability to bear bending and some degree of shock. Here, again, it is necessary to bear in mind the object of the test, and so to devise the best means for attaining the end in view, and not something more or different. A steam cylinder is subject to shock from the sudden admission and emission of steam; the H.P. cylinder walls are always in tension, varying from a maximum at cut off to the minimum at commencement of exhaust, the metal will be at a temperature some what but not much below that of the steam at admission; the stress on the material must not be high—not high enough at anyrate to produce perceptible stretch, or there will be an increase in diameter of bore and a consequent leakage past the piston. Moreover, the metal must be such that, while capable of being bored and machined, it must have the quality which admits of taking a polish when rubbed, and that without much lubricant, and must in many cases continue to be rubbed without a lubricant at high temperatures without abrasion of the surface.

Such a metal must have, therefore, qualities that no tensile testing machine can prove to exist; in fact, high tensile material is desirable rather for the water tests put on the cylinder than those applied in every-day work. Cross-bending of a bar freely supported is still the most satisfactory method of proving its strength and capacity for taking a sudden load—that is, withstanding shock. Chemical analysis is perhaps the safest test for making sure of its other virtues—in fact, the successful iron moulder to-day establishes his reputation by the care with which he selects his irons for mixing, so that the chemical composition shall be definite, and one known to be co-existent with high mechanical properties. But since the most important parts of an engine which are made of cast iron always work when at a high temperature, mere cold tests are not sufficient; such tests as are made should be with the metal at a temperature 5 to 10 per cent. higher than that at which they are exposed when at work. These remarks apply with even greater force to the bronzes, some of which, while being very strong at 160° F., are weak and unreliable at 420°, while at 600° are of no use whatever.

It is usual to test by water pressure the more important parts of a marine engine before placing them in the ship. Here, again, it is not clear, judging by results, what is aimed at by the authorities in making such tests. Primarily safety is the first consideration, and was probably the only one originally, when the ability to calculate strains and stresses was limited, and the experience to guide the designer equally limited. To-day the water test prescribed for cylinders appears to be with the object of proving the strength of the casting, and incidentally its soundness. Unfortunately, these two things are often confounded by inspectors and surveyors, so that a casting which is quite strong enough and even sound enough for its work is condemned because at *test* pressure there has been a leakage through local sponginess although at *working* pressure there was no leakage. The same test for boilers is open to the same misunderstanding, for it is not unknown to find parts of a boiler which at the pressure at which the safety-valves are to be set are unaffected, do actually bulge and deform, so as sometimes to produce leakage at the test pressure, especially when that test is at double the working pressure. Tightness is one thing, and a very desirable thing, but it should be guaranteed under working conditions. Therefore, that a casting leaks on testing with cold water at a pressure far beyond that at which it works under steam is no proof of unfitness for its work. The high pressure proves its strength; its ability to sustain steam without loss can only be proved with steam.

With a complicated casting like a cylinder it is necessary to prove the following, viz., that—

(*a*) It is strong enough to sustain the loads, shocks, and thermal changes coming on it internally and externally.
(*b*) Under working pressure there is no deformation of its body or adjuncts that will interfere with the good working of the piston and valves.
(*c*) Under working pressure there is no leakage which will cause any loss of steam by escape to the air or exhaust.
(*d*) And the metal is such as can be machined satisfactorily, and leave a good working surface for the rub of pistons, valves, etc.

The low-pressure cylinder differs from the others, inasmuch as at exhaust the internal pressure is much less than that external, and consequently that the metal is during that period in a state of compression, and the body exposed to collapse. There is no test ever applied to prove its ability to withstand this state, so that the water test is limited to a question of tightness rather than strength.

The casings and stators of turbines must be strong enough to safely sustain the internal pressure of the steam, which may be that of the boilers in a complete turbine, as it is in the high-pressure portion of a compound one. In actual work it is only at the entry of the H.P. end that is exposed to full steam pressure, but in case of a hitch it might extend throughout. But in the Parsons turbine there must be no increase in diameter when working under pressure which would materially increase the clearance and the loss due to it, nor must the body change from its true circular section; therefore, when it is tested it should be carefully gauged when at working pressure, as the cylinders of reciprocating engines require doing.

The L.P. portion of a turbine case is liable to an absence of pressure throughout its length when standing, and, therefore, like the L.P. cylinder, is subject to collapsing forces, its ability to resist which is never tested, and only guessed at by testing it under internal pressure high enough to prove the soundness of the casting.

The testing of a condenser body is equally unsatisfactory for the same reason, for in designing it to resist such a test material is added, and the form adopted which is quite useless to help it against working strains. If the stiffening ribs of a casting are on the right side for test purposes, they are on the wrong one for working loads, and *vice versâ.* The waterways and ends of a condenser are subject to slight internal pressures, against which provision can be made without such objection, as anything added for testing really helps these parts to sustain the latter rather than working loads. The British Admiralty require the following water tests carrie.. out satisfactorily:—

Admiralty Rules for Testing Machinery.

615. **Water Pressure Tests Generally.**—The water pressure tests required are to be carried out after the parts are machined, and before they are painted or lagged, or covering of any kind is placed on them.

616. **Tests before Fitting on Board.**—The boilers, turbines, and their connections, condensers, air pumps, and all filter, drain, receiving, lime, drip, and oil tanks, and the feed tanks, when not built into the ship, are to be tested by water to the required pressures before being put on board the ship; the steam and exhaust pipes, all boiler mountings and connections, and other important castings are to be tested before being fixed in their working position. The tests for the turbine cases other than that after completion are to be made after grooving, but before the final cut is taken over them.

The turbine cases are to be tested in the shop after blading for several hours under steam pressure, and they are to stand this test and all water-pressure tests without any distortion which may be objectionable.

617. **Tests after Fitting on Board.**—After the machinery is fitted, complete, on board the ship, a water test is to be made of all steam and exhaust pipes, feed pipes, boilers, condensers, and air pumps, with the pipes and connections of the preceding parts, to the pressures mentioned, and the turbine casings to such pressures as may be approved; this test is to be applied to the main and auxiliary machinery, air reservoirs, air service and oil fuel pipes and fittings generally. For this rest the lagging may be in place on all boilers and castings, but all pipes and all joints are to be uncovered. The steam pipes are to be tested separately to twice the full boiler pressure. They should be tested up to the full boiler pressure with all important valves shut, valves which may be subject to pressure on either side having the

pressure applied independently on either side. In continuing the test up to the maximum test pressure specified, the valves should be left open.

Steel steam pipes may be lagged for water test on board; but the flanges are to be left uncovered.

Note.—During the test of the oil fuel suction pipes the strainer box covers are to be removed to prevent the possibility of the double bottom compartments being subjected to pressure.

618. **Particulars of Tests for Various Parts.**—The various parts, as well as the corresponding spare gear, are to be tested by water to the following pressures per square inch:—

Air reservoir bottles, separator and charging columns and air service piping, with their spare gear, and the parts of the pumps, etc., subjected to the maximum air pressure,	4,000 lbs.
Each solid-drawn boiler tube and oil fuel heater tube of and below 2 inches external diameter,	2,500 ,,
Each solid-drawn boiler tube above 2 inches external diameter,	1,500 ,,
Each condenser, distiller, and oil cooler tube,	1,000 ,,
The feed pump chambers, feed discharge pipes, valves, and any apparatus subject to feed pump pressure, and main steam pipes if of welded wrought iron,	Three times the full boiler pressure.
The water chambers of ash ejector pumps, discharge pipes, valves, etc., up to and including the shut off cock on hopper, if so fitted,	600 lbs., unless otherwise ordered.
Blown-out pipes and valves, the main steam pipes, stop valves, strainers, and all fitting up to the high pressure ahead, cruising, and high-pressure astern turbines; the auxiliary steam pipes, stop valves, separators, steam side of oil fuel heaters, all boiler mountings and castings, and all fittings which are subject to the full boiler pressure,	Twice the full boiler pressure.
Boiler-room oil fuel pump barrels, with suction and delivery chambers, suction and delivery pipes, and valves (excepting sluice valve and strainer on inner bottom), filters, and oil side of heaters,	400 lbs.
The boilers when erected in the shop, after fitting on board, and in any case before raising steam,	One and one-half times the full boiler pressure
Cylinders and all parts of auxiliary engines and evaporators, subject to boiler pressure,	
The suction passage of the feed pumps,	300 lbs.
The low-pressure cylinders of compound auxiliary engines,	235 ,,
The pump ends, discharge valve-boxes, and air vessels of the fire and bilge engines, excepting those used in connection with See's ash ejectors (if fitted),	250 ,,
High-pressure ahead turbine:—	
Steam end and inlet cover,	255 ,,
The remainder,	200 ,,
After completion,	170 ,,
Cruising turbine:—	
Steam end and inlet cover,	255 ,,
The remainder,	200 ,,
After completion,	170 ,,
High-pressure astern turbine:—	
Cylinder and inlet cover,	255 ,,
Exhaust cover, if separate,	200 ,,
After completion,	170 ,,
Low-pressure turbine:—	
Astern cylinders and inlet covers,	50 ,,
Ahead cylinders and inlet covers,	30 ,,
Exhaust chamber and passages,	30 ,,
After completion,	30 ,,

Receiver pipe and valve between cruising and high-pressure ahead turbine,	255 lbs.
Feed suction pipes,	250 ,,
Air compressing chambers of compressor for cleaning boiler tubes,	150 ,,
Air service piping for cleaning boiler tubes,	100 ,,
The pump chambers of distilling, forced lubrication, and engine-room oil fuel pumps, and lime tanks,	100 ,,
All underwater valves up to and including 12 inches diameter attached to the skin of the ship or shipbuilders' tubes, excepting the boiler blow-out valves, on the working side of the valve with the valve closed, and all pipes except where otherwise specified,	100 ,,
Receiver pipes between the high-pressure ahead and low-pressure turbines,	60 ,,
The air pumps,	60 ,,
The auxiliary exhaust pipes, evaporator cases, and the bilge suction pipes,	60 ,,
Sluice valves and strainers in oil fuel suction pipes immediately above inner bottom,	60 ,,
Receiver pipes between high-pressure astern and the low-pressure astern turbine,	50 ,,
The working side of all underwater valves above 12 inches diameter, with the valve closed,	50 ,,
The air pump pipes and connections,	50 ,,
Lubricating oil suction and drain pipes,	50 ,,
The main eduction pipes,	30 ,,
Grease extractors—As may be ordered, but not less than	30 ,,
Hoppers of See's ash ejectors (if fitted) on maker's premises,	30 ,,
The steam chambers of the condensers, the auxiliary engine drain tanks if fitted, the steam chambers of distillers, and distiller cases,	30 ,,
The shaft casings *before* being fitted on the shafts,	30 ,,
The shaft casings *after* being fitted on the shafts (This *test* to be made with *boiled linseed oil.*)	30 ,,
The water chambers of the condensers, the circulating pump cases, all valves and pipes between the corresponding sea inlet and outlet valves,	25 ,,
The feed, distiller test, drip, and oil tanks,	10 ,,
See's ash ejector (if fitted) and piping complete when erected in place on board,	To a head of 30 feet of water above the hopper.

620. **Valves and Valve Boxes.**—All steam stop valve boxes are to be tested to the full specified water pressure with the valves open and also with the valves closed, when they must be tight on their seats. A subsequent test of these valve boxes is to be made under steam of the specified working pressure, with the valves closed, and the pressure on the working side of the valves. These tests are to be carried out in the shops before the valves are fitted on board the ship, to show that the valves are steam tight. The efficiency of the opening and closing gear is to be tried during these tests. Underwater valve boxes are to be tested as a whole with the valves open, and are also to be tested above the valve with the valve closed to the full specified water pressure.

621. **Steam Tests.**—All high-pressure pistons and slide valves of the auxiliary machinery, all reducing valves, are to be tested under steam on the Engineer's works to the satisfaction of the Overseer before they are fitted on board ship.

623. **Tests of Springs of Safety Valves.**—Before the steam trials, all the safety valves and their springs are to be taken out an ' replaced for examination, measurement, and test, to ascertain the exact load on the valves, and that they are quite free in their seats; in the case of the boiler safety valves, this test is to be supplemented by raising steam in the boiler to full pressure, the springs being ajudsted while the full pressure is maintained in the boilers. The stops are then to be fitted; after which, the accumulation test of the valves is to be made.

624. **Tests of Gauges.**—Before proceeding on the steam trials the pressure, vacuum, and compound gauges are to be removed, tested, and replaced by the Engineers.

The Italian Government require the following tests to be made in the case of machinery fitted in vessels registered in that country. W.P. is the working pressure in the boilers:—

(*a*) High-pressure or 1st cylinders and valve chests, where W.P. is less than 142 lbs. per square inch are to be tested to 1·5 W.P., and when W.P. is over 142 lbs. to (W.P. + 71 lbs.).

(*b*) Second cylinders and valve chests are to be tested to W.P.
(*c*) Third ,, ,, ,, 0·66 W.P.
(*d*) Fourth ,, ,, ,, 0·33 W.P.

If the receivers are fitted with safety valves sufficiently large, and loaded to the maximum W.P. of each cylinder, the test may be limited to twice the loa l pressure.

(*e*) The steam jackets of all cylinders to twice the pressure they are subjected to in working condition. If reducing valves are fitted, safety valves must be fitted on the reduced pressure en '.

(*f*) H.P. casing or cylinder of a turbine to 1·33 W.P.
(*g*) Astern-going turbine casing to W.P.
(*h*) L.P. turbine, admission end, 0·33 W.P.
(*i*) ,, exhaust end, 28·5 lbs. per square inch.
(*j*) Built condenser and all condensers for turbines are to be tested to 28½ lbs., while for cast-iron condensers for reciprocators 21 lbs. is sufficient.
(*k*) Safety valves and feed pumps to 2 W.P.
(*l*) Air and circulating pumps to 28½ lbs.
(*m*) When all the connections are made on board and fitted complete, a trial under steam in the usual way.

The British Corporation require the following water tests:—

(*a*) All evaporators and high-pressure feed heaters tested satisfactorily at maker's works.
(*b*) Feed-water filters between pumps and boilers to 2·2 W.P.
(*c*) All auxiliary steam pipes exceeding 4 inches in diameter, and all main and refrigerator machine steam pipes, not less than 2 W.P. when of copper, and 3 W.P. when of iron or steel.
(*d*) All boiler feed pipes 20 per cent. higher than required for steam pipes —that is, 2·2 W.P.

Lloyd's Register of Shipping require no water testing, except on the boilers, etc., which are, as already stated—viz., the test pressure is 1·5 the W.P. plus 50 lbs., and on pipes twice W.P. for copper, and three times for steel.

The Board of Trade also requires very little water testing beyond that of the boilers and their pipes. H.P. turbine cases to 1·33 W.P., L.P. condenser end 30 lbs and the astern-going case to W.P.

Tests required by the Rules of the Germanischer Lloyds.—The boilers must be tested by hydraulic pressure 1·5 W.P., if the working pressure is not more than 142 lbs.; above that pressure it need be only (W.P. + 71 lbs.), where W.P. is the working pressure above atmospheric. Main steam pipes, to twice W.P.

H.P.	cylinders of	all engines to (W.P. + 71 lbs.).	
L.P.	,,	a compound engine to	42·7 lbs.
L.P.	,,	a triple- and quadruple-compound engine to	28·5 ,,
M.P.	,,	a triple-compound engine to . .	0·66 W.P.
1st M.P.	,,	a quadruple-compound engine to . .	0·75 ,,
2nd M.P.	,,	,, ,, ,, . .	0·40 ,,

Trials under Steam are made for various purposes; some are of the simplest kind, others are long, variable, and costly. In any case each trial should have a definite object or objects, and when there are more than one they should not be confused, so that one or more may be obscured. As a rule, the primary object common to all concerned is to demonstrate the actual and satisfactory fulfilment of a contract, whereby the engine builder has undertaken to deliver machinery capable of performing certain functions satisfactorily. The following are the principal things involved in such a contract as is usual:—

(1) That the machinery is actually as described in the contract and specification attached thereto, and accords with such plans as have been submitted and approved.

(2) That it is approved by and passed by the Surveyors of the Board of Trade, Lloyd's, or other body, where sanction is necessary for the ship to have the status intended by her purchaser.

(3) That the engines will work satisfactorily at the full working speed with only such attention and lubrication as is usual in similar ships on service, and that they can be depended on to do so for a reasonable period.

(4) That the boilers will generate and supply a sufficient quantity of dry steam at a pressure 1 per cent. less than the load on safety valves for the above purpose, and for domestic and other needs, as designed.

(5) That the consumption of steam by the engines is not in excess of that usual with similar machinery, and that the expenditure of fuel in the boilers is also as low as usual. In other words, that the efficiency of both boilers and engines is what it should be.

(6) That the power developed is in accordance with the contract, and is obtained on conditions conducive to good and economic working.

(7) That the engines neither vibrate more than such engine should when properly balanced, nor set up vibration in the hull of the ship.

(8) That they are easily under control, and can be started, stopped, and reversed without fail in quite a quick way.

(9) That the condenser and its connections are all absolutely air-tight and capable of maintaining a vacuum within 2 inches of the barometer when the sea-water is not above 75° F. without an excessive quantity of cooling water, and with the hot-well temperature not less than 100° F.

(10) That the hot surfaces of boilers, pipes, cylinders, and all fittings are properly and securely covered with a non-conducting material, so that the temperature at the surface of the lagging or covering is not more than 120° F., nor 25° F. above that of the atmosphere, and generally that there is no loss of heat by conduction or radiation that can be avoided by reasonable and practicable means.

(11) That the ventilation of the engine-room, stokeholds, and all other places where attendants have to keep watch or remain in for more than 10 minutes at a time shall be such that there is a plentiful supply of fresh air, and the temperature does not exceed that of the external air by 20° F. nor above 115° F.

(12) That the means for access to and escape from all such parts is practically free from risk to life or limb, and can be effected in a reasonably short time in case of emergency.

(13) That all such places are or can be sufficiently lighted without notice or loss of time to permit of the attendants performing their duties efficiently and quickly.

(14) That the fire appliances are conveniently placed and accessible at all times, so as not to be beyond use in case of fire in their neighbourhood, and that they are efficient for the service desired from them.

(15) That in case of storms or heavy seas when there may be necessity for "battening down," there is sufficient ventilation and means of access and egress in machinery spaces for the well-being of the staff on watch.

(16) That the pumping arrangements generally, and especially those affecting the safety of the ship, such as the bilge pumps, pipes, and fittings, are satisfactory in size and position, accessible and easy to examine and clean, and maintain generally in an efficient state.

(17) After the trials the boilers should be carefully examined to find if there are any leaks or tendency to leak at any place, and if there have been any general or local straining due to heat and pressure.

The machinery should be inspected with equal care to ascertain if there be any signs of weakness or inefficiency in any part, or such indications of wear or tendency to wear at the guides, slides, journals, pins, valve motion, etc., or any need of modifying the adjustments of any of the working parts, or necessity to improve the lubricating arrangements.

(18) The auxiliary machinery should be tested to prove that the capacity and efficiency of each and every part is such as specified, and necessary for the good working of the ship, and that they may be depended on to fulfil their several duties on a prolonged voyage at least 25 per cent. longer than usual on the service intended.

(19) That the arrangements for taking in fuel and supplying it to the boilers are satisfactory, and that the bunkers are ventilated in such a way as to preclude the possibility of gas accumulation with explosions.

So much for the trials of the machinery and the inspection of it afterwards, so far as the manufacturer is concerned; but it is only after all a means to an end, which is the propulsion of the ship at a certain speed in the most efficient way possible—that is, to enable the highest speed to be attained with the least expenditure of fuel, stores, wear and tear, and attention. The saving of a fraction of a pound of fuel per I.H.P. per hour is a desirable thing.

but it may be purchased dearly if there is a material increase in consumption of oil, in the repair bill or in the staff charges.

The Admiralty Ship Trials up to thirty years ago were quite simple and inexpensive, and it may be added by modern standards equally unsatisfactory, chiefly on account of their incompleteness. The engines were tried at moorings and moved at moderate speeds for a few hours to give everything a "rub down." After the adjustments found to be necessary from such a trial, the ship was taken to sea for a preliminary run, and if full power was developed for even a short time it was deemed sufficiently satisfactory to warrant the attempt of the official trial two days after. This official trial was with the object of demonstrating the ability to attain rather than to maintain the I.H.P. contracted for, and incidentally to find the speed of the ship by making six runs on the measured mile, during which the revolutions were determined by mechanical counters, sets of diagrams taken from the cylinders by tested indicators, and the pressure of steam at boilers and engines noted, as also the vacuum in condenser. If the six runs, half with and half against the tide were performed satisfactorily, steam was allowed to drop until half the boilers could be shut off without blowing of the safety valves. The engines were then reduced in speed by notching up the valve links or putting expansive valves in gear, so that full pressure could be just maintained on the half set of boilers. Four runs on the measured mile were taken under these circumstances on the same conditions as before as to observations. The trial was then practically over, especially if there were no novelties requiring special tests. Sometimes there would be a run afterwards to let the pressure down, as there would be a run preliminary to the measured mile to get the pressure up, and generally to "tune up," as it is now called, the machinery. The mean or true speed of the ship was ascertained as it is to-day by taking a mean of means and the I.H.P., steam pressure, revolutions, vacuum, etc., by taking the mean of the records made on the measured mile of 6,080 feet. If the bearings had run fairly cool without the application of the hose, the vacuum was within 4 inches of the barometer, and the I.H.P. not materially less than contracted for the trial was satisfactory, so far as the engineers were concerned.

The Trials of the Ordinary Merchant Ship were then pretty much the same, and to-day those of a cargo boat are no more elaborate and often even less so. In fact, the chief object of such trials now is to satisfy the owners and the staff that the machinery is to be relied on to work satisfactorily in its primary duty of causing the transport of the ship from port to port without exceeding the daily consumption of fuel promised by the builders. If the speed talked of was 10 knots, there is not any question raised if the speed is 9·8 or 10·2 actually attained.

The Modern Steamship Trial, as introduced by the late William Denny, of Dumbarton, is quite a different affair, and is probably due to the unique facilities for such provided by nature in the Clyde estuary, where there is ample room for the trials of the largest and fastest ships in deep water with very little current, and sheltered from wind and waves, all within easy reach of the shipbuilding yards and docks.

Progressive Trials, as made by Mr. Denny, and now by all builders of ships whose speed and efficiency are of importance, are comparatively simple, and involve little, if any, more cost than the old Admiralty ones. Incidentally these latter gave the two speeds and sets of facts appertaining to them: and as at times the Admiralty wanted to know what a ship could do with a quarter or a third of the boilers in action the trials thus made had the *desiderata* of Mr. Denny's proposals—viz., three speeds with their facts and figures were noted, and with them curves of I.H.P., revolutions, slip, and efficiency could be plotted with the speeds as base lines. This was what Mr. Denny showed was necessary to demonstrate the efficiency of the ship under trial, and guide the designer in dealing with the future problems. It also showed most clearly and unmistakably the efficiency of the machinery, as well as that of the ship, and perhaps what was of equal and possibly greater importance, the efficiency or otherwise of the screw propeller.

It is, therefore, desirable that all ships should be tried at three differing speeds always, and when time and circumstances permit at four, in order that there may be no question as to the trend of the different curves. If such problems as arose on the effect of the depth of water on the speed and power of a ship, even a greater number of trials are necessary. If three only can be made, they should be at full power, at 78 per cent. of full power revolutions, and at 40 per cent. If four trials can be done, they should be at 25 per cent., 50 per cent., and 80 per cent. If a set of runs can be made satisfactorily at a lower speed than 25 per cent., so much the better, but to do so there should be practically no current or wind, and great care taken to maintain both steam and vacuum with the slightest possible variation. The revolutions should be counted mechanically during the whole time the ship is on the mile, and for the exact time if possible, so that if the slip is constant there should be always the same number of revolutions per mile at every speed. If the engine is fitted with a speed indicator, it should be watched for variation, especially during the full speed runs for thereby cavitation so called, or tendency to race from any cause, will be noted; in fact, it is always desirable to have an automatic recording revolution indicator, which would show these variations and their magnitude.

Progressive Trials at Sea can be taken quite easily in such a way as will enable those interested in checking the efficiency of ship and machinery from time to time. Such trials should be made, because they are really of commercial advantage, on all important oceangoing steamships, and may be carried out at trifling expense by means of a good self-recording patent log kept for the purpose, so that it may be looked on as a standard. Its error can be easily found and noted, and so long as it is always used it will not matter as to the amount if it is constant. The ship should run for twenty minutes, or longer if possible, at three differing rates of revolution, which do not vary during each trial. Indicator diagrams are carefully taken, or the torsion meter is put into requisition. Revolutions are counted, steam pressure and vacuum noted, as on the official trial.

It need hardly be said that a series of trials such as these should be made in comparatively smooth water with the wind steady and as low as possible, and that either in deep water or water whose depth does not vary much. The time occupied need not be more than $1\frac{1}{2}$ hours, and the loss of distance run will be so inconsiderable as to be no bar to the performance of such a useful trial. From the data obtained a set of curves can be plotted in the usual way, and compared with those furnished by the builders as the results of the trials at the measured mile, or with those made by the ship's staff when in known thorough good order, with the bottom clean and fresh painted.

Endurance Trials are now deemed to be as necessary as the speed trials for all important ships. The British and most foreign naval authorities require all ships to endure satisfactorily prolonged trials at voyage and cruising speeds, as well as at full power. during which careful observation is made of the consumption of fuel as well as the consumption and waste of water; the performance of all auxiliary engines and appliances are as carefully noted as that of the main machinery, and altogether the trials are of an exhaustive as well as exhausting nature for all concerned, and very different in every way from the old speed trial trips of last century. Destroyers and torpedo boats had formerly a prolonged run at 10 knots, during which the coal was carefully measured and the ship's radius of action gauged by the results. Latterly 14 knots is the usual speed for craft of this kind to be tried at for this purpose.

Larger craft, including all cruisers, are subjected to a series of trials after having had a basin or dock trial and a preliminary trial at varying speeds to enable adjustments to be made by the contractors.

(*a*) The first trial is of 30 hours' duration, with the engines running so as to develop about $\frac{1}{5}$ the horse-power; during this trial, as at all others, the amount of waste of water is noted by measuring the make up supplied by the evaporators, and it is limited to 5 tons per 24 hours for each 1,000 H.P. developed. The coal is carefully weighed, and the total consumption of water estimated from data obtainable. During this trial there are tests made of the ability of the cruising turbines when so fitted, each lasting about three hours.

(*b*) The second trial is also of 30 hours' duration, of which the first eight is with the engines developing about four-fifths the full power, and the 22 hours remaining at about three-fifths. During this trial the water waste must not exceed 4 tons per 24 hours for each 1,000 H.P.

(*c*) The third trial is at full power for eight consecutive hours, when the waste must not exceed $3\frac{1}{2}$ tons per 24 hours per 1,000 H.P.

(*d*) There are then various short trials to demonstrate the ability of the machinery to respond quickly and efficiently to the orders from on deck. To find what is the speed of the ship going astern when using half the boilers as well as all of them. To note the time taken after the order is given to stop the ship to bring her to a standstill. Also, the time taken in starting at rest to attaining the maximum stern-going speed.

The pumping capacity and other duties of auxilaries.

(*e*) The engines are then opened up throughout and carefully examined, the condensers tested by water, and after closing up again and adjusting.

(*f*) A final trial of 24 hours' duration, of which 12 must be a continuous run at half power and 12 hours at such powers and revolutions as may be directed by the Admiralty, with all the boilers alight ready for use, if not actually in use during the whole time.

The allowance of extra feed above stated as about 2·33 per cent. of the total consumption of the water of the main engines at the time.

Such trials as these are very crucial and very costly, the fuel alone will cost £1,300 to £1,500* with a cruiser of 25,000 I.H.P., and the wages and keep of the staff together with the stores a further large sum; but they

* At pre-war rates for S. Wales best coal.

have been found necessary, and are continued to avoid the possibility of a ship being taken over from contractors with latent defects which only by a prolonged trial, such as would take place on service, would be made manifest.

In the Mercantile Marine no such crucial trials are deemed necessary, except in the case of large and important ships having machinery novel in design, size, or type, and intended for a long or important voyage service, where break-down or even temporary stoppage would be fatal to her character and damaging to the service and owner. There is, however, beyond all this the interest of the builders of both ship and machinery to consider; their character and reputation are equally involved in the success or failure of such ships, and they, therefore, prefer, or should do so, these drastic trials which are nowadays insisted on. In the case of cross-channel steamers a trial trip of 100 miles made at full speed without a stoppage on service conditions is considered sufficient when the route on which she is to run does not exceed that distance. Before commencing this endurance test, it is requisite and usual to have progressive measured mile trials, so that by carefully recording the revolutions the actual speed can be checked against that given by the patent logs, or that estimated by the navigating staff from observation.

Since turbines have been employed to drive marine propellers, a new set of trials has been devised to prove their ability to manœuvre the ship. For this purpose it is usual to note how long after the order is given to stop and reverse the engines the ship comes to rest, and the distance run. This is done at two or three rates of speed, as is also the turning of the ship by moving one wing engine ahead and the other astern. The introduction by Mr. Blenkinsop of his special patented reversing gear, whereby the middle engine under these circumstances is always running ahead has shown that the ship turns much quicker than when the centre screw is at rest. Perhaps the reversion to twin screws from the triple and quadruple ones is partly due to the inability of such small screws to quickly control the motion of the ship, as well as to the advantages derived by geared turbines.

The machinery of express passenger steamers is usually opened up after an endurance trial, but not to the same extent as in the naval ships; indeed it is only where trouble has been experienced or suspected that a careful examination is deemed necessary, except that the cylinders and main valves are in any case examined, as also the internal parts of the boilers, otherwise it depends very much on the custom of the superintending engineer how much more will be distributed for his inspection. It was the custom formerly to permit of the water service being in action throughout a trial, rather as a precautionary measure than a remedial one. To-day trials should be, and generally are, run without the application of any water, and no engine can be deemed satisfactory that will not run without heating of bearings, pins, or guides when lubricated only; it may be that the oil is bad or that the lubricating arrangements are faulty, these defects can be remedied or at least excused and put right as soon as possible; lack of bearing surface, poor quality of white metal, or faulty finish to journal and brasses cannot be treated lightly. Forced lubrication, however, should be followed as the most successful method for avoiding trouble and ensuring efficiency.

APPENDIX A.

The Diesel Oil Engine.

The Diesel Type of Oil Engine is, no doubt, the best of the oil engines for use on board ship, inasmuch as it works on a small consumption of a safe and comparatively cheap fuel, so that it is economic in working, and premits of a large radius of action on a given weight of fuel; the engine-room staff is no larger than required with steam engines, and there is no boiler-room crew at all. Moreover, so far, it is the only internal combustion engine that has been subjected to exhaustive trials at sea in large sizes.* The space occupied by an oil engine installation is considerably less than that of a steam engine one, but there is no corresponding saving in weight.

Single-acting 3-cylinder 4-cycle heavy type slow-running Diesel engines, 250 to 700 B.H.P., weigh 460 lbs. per B.H.P.

Single-acting 4-cylinder 4-cycle heavy type slow-running Diesel engines, 300 to 900 B.H.P., were 400 to 435 lbs., while the 2,000 H.P. modern are 265.

Single-acting 4- to 6-cylinder 2-cycle marine type at 200 revolutions per minute, 150 to 224 lbs. per I.H.P. The larger at 150 revolutions, 140 lbs.

Single-acting 4- to 6-cylinder 2-cycle marine high-speed engines, 90 to 100 lbs. per I.H.P.

The very lightest marine Diesel engines, as made in Germany for submarines and torpedo boats, are 35 to 40 lbs. per I.H.P., which is about the same as that of the steam turbine installations in destroyers, including boilers, etc. British machinery in "*oilers*" is 50 to 56 lbs. per S.H.P.

The Efficiency of the Diesel Engine is not so high as that of steam engines, *qua* engine, for the thermal is 42 to 48 per cent. at most, and the mechanical about 84 per cent. at full power, and at dead slow only 33·4, so that in comparing it with a turbine or steam reciprocator it is not sufficient to take the consumption per I.H.P. as the criterion. Moreover, the ratio of maximum to mean pressure (5 to 6) is so great, and as there is only one impulse per revolution with the two-cycle engine, the torque is very irregular; consequently the efficiency of the screw driven by it must be less than when driven by a steam reciprocator, and much less than with a turbine.

But to compare the efficiency of the whole installation of an oil plant with that of a steam it is necessary, of course, to take into account the efficiency of the boilers. For this purpose consumption of fuel is the criterion, and efficiency is arrived at by comparing the output with the theoretical possibilities of the fuel.

The heat equivalent of a horse-power per hour is—

$$\frac{33{,}000}{778} \times 60 = 2{,}547 \text{ B.T.U.}$$

The marine Diesel engine consumes 0·4 lb. of fuel per S.H.P., the thermal value being 18,500 B.T.U., the S.H.P. requires 7,400 per hour. The possible H.P. with this quantity is 7,400 ÷ 2,547, or 2·905; the efficiency, therefore,

* The so-called semi-Diesel, but really a low-pressure, heavy oil engine with ignition attachments is now very popular, and made in larger sizes than formerly.

of such an engine is 34·4. By the same test a modern turbine consuming, with its auxiliaries 10·5 lbs. of steam requiring 0·7 lb. of oil fuel per S.H.P. hour will have an efficiency of 19·7; and a quadruple reciprocator worked with high superheated steam, about 17 per cent., or only half that of the oil engine.

The Rate of Revolution of this Oil Engine is, as a rule, higher than that of a steam engine; in fact, it must be high to be efficient and trustworthy with a limited number of cylinders; but it is, of course, considerably less than that of the turbine. For standard light engines up to 500 H.P. about 500 revolutions is a common speed; up to 1,200 H.P. they are about 400. The ordinary heavy engine of the mercantile marine is run at 275 to 300 revolutions per minute up to 500 H.P., 215 to 260 up to 1,200 H.P., the large engines in ocean-going craft 150 revolutions on trial conditions is usual, and in the very largest, as in fig. 325, 100 to 125 revolutions. Naval oil engines as in small craft are usually run at about 400 revolutions, the piston speed being 960 feet per minute, while the piston speed of large merchant ships is 750 to 850 feet.

Every Diesel Engine must have an air-compressing pump of some kind to provide the air for pulverising and injecting the fuel into the cylinder, whose contents have been compressed to 500 or 600 lbs., and even higher in some cases, where the fuel has a high flash point, such as coal-tar oil, which has 400° F., whereas that of the ordinary residues is about 200° F. Sometimes these pumps are driven by an independent steam engine or electric motor, when steam or electricity is available for the purpose; latterly, especially for marine purposes, the pump is driven from a crank at the fore end of the main engine. The quadruple *Reavell* is a favourite one for this purpose, but often there is only an ordinary two-stage pump, as shown in the illustration (fig. 320).

The Two-cycle Engine is bound to have a low-pressure pump to supply the air necessary to scavenge the oil cylinder and fill it with fresh air prior to compression. These pumps are quite simple, and not unlike the old air pump of the surface condenser, and may be, and are, worked in the different ways, as found convenient formerly for them. They may be worked by levers, by a crank or cranks, or direct from the piston or crosshead of the engine itself (*v.* fig. 323). The English engineers as a rule prefer to work these pumps with levers, etc. Sulzers have adopted the plan of having the pump in line with the oil cylinders, and driving it from a crank below it. The Nuremberg, Augsburg Company drive direct from the oil trunk piston, as shown in fig. 321, where the pump chamber is immediately below the oil cylinder, and the ratio of their diameters 1·4 to 1·6, so that the capacity permits of a 25 per cent. excess of air to be delivered. By this latter arrangement the oil cylinder trunk is exposed to the cooling action of the air, and thereby prevented from getting overheated, and, on the other hand, the air is somewhat heated before entering the oil cylinder. The air-pump trunk in this case contains the connecting-rod end and its gudgeon pin, so that it is really the guide, and takes the thrust of the connecting-rod; it is, however, cool, and the space in it large enough to permit of an ample surface of gudgeon and means for efficient lubrication, which is not the case when the ordinary piston trunk of the oil cylinder, as in fig. 49. The fuel consumption of the two cycle is 6 per cent. higher than that of the four.

In the more Modern Designs of most makers of these engines there is a trunk, with a piston-rod, and its crosshead, slipper guide, etc., as found in the

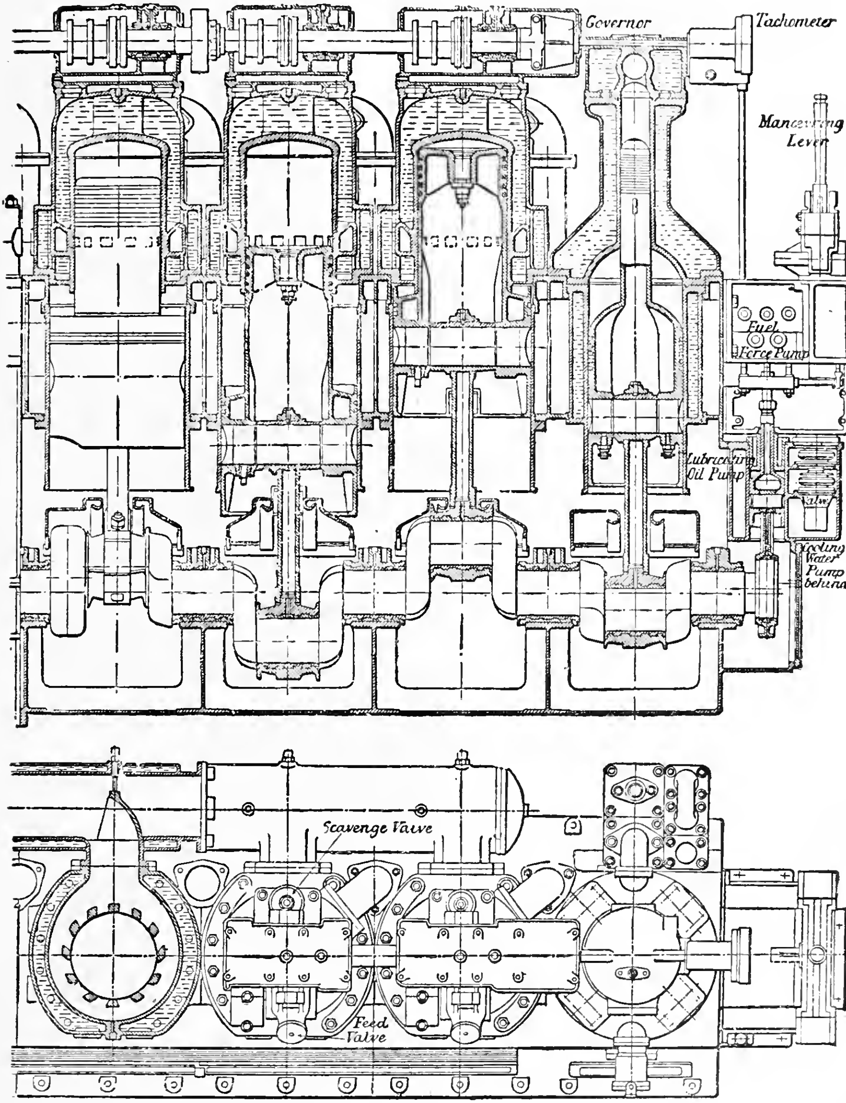


Fig. 320.—Two-cycle Single-acting Marine Engine for Heavy Oil—Diesel System. (German Design.)

steam engine (*v.* figs. 324-5). The engine is two-cycle, single-acting, and the oil cylinder quite open at the bottom, so as to be subject to the cooling action of the ventilating air. All large marine engines, whether single- or double-acting, should be of this design, so that the connecting-rod end and its gudgeon are free from the surrounding trunk, and in view, where they can be examined from time to time, as well as be efficiently lubricated.

The Double-acting Oil Engine has not come yet, and although on the Continent it has been tried on quite a large scale, there is reason to believe it has not been altogether satisfactory. If it is on the two-cycle system

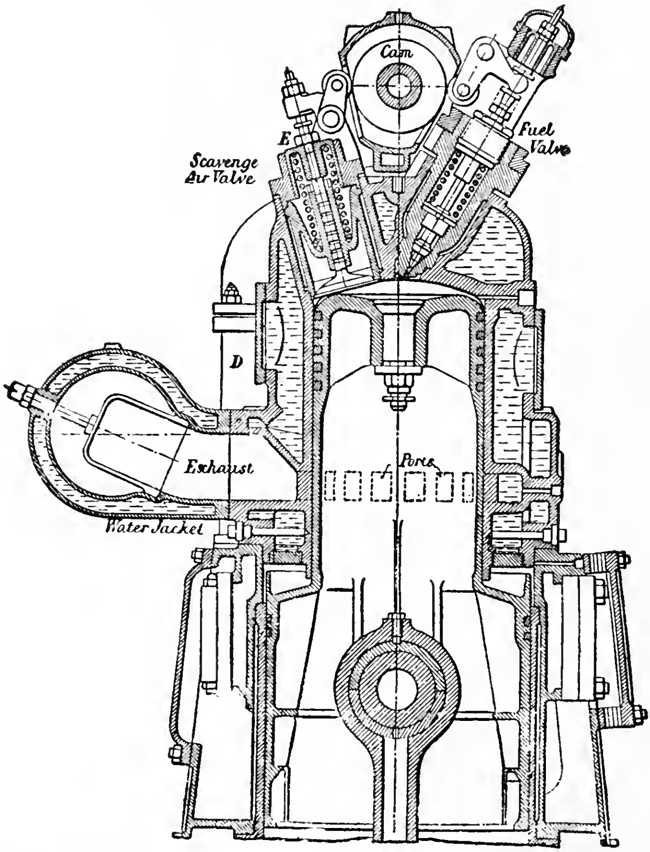


Fig. 321.—Cylinder and Air Pump of Two-cycle Single-acting Diesel Engine (German Design.)

the cylinder is necessarily of great length, and the engine itself so high (or long if horizontal) as to be prohibitive on most ships. It is also almost an impossible thing for an engine working at such high temperatures as the Diesel to keep reasonably cool with an explosion or conflagration at each stroke, even with water-jackets and water-cooled pistons; then, too, there are quite serious difficulties with the piston-rod glands even on the double-acting four-cycle system, which must be magnified when the two-cycle is adopted. The mean pressure in a Diesel two-cycle engine is usually 100 to 110 lbs. per square inch, and may be in some cases as high as 125 lbs.; the referred mean pressure of a quadruple reciprocating steam engine is 40 to 50 lbs., or less than half that of the oil engine; but the pressure in their L.P. cylinder is only 10 to 12 lbs., so that the size of the oil cylinders may be quite small compared with those of the steam ones, in spite of being single-

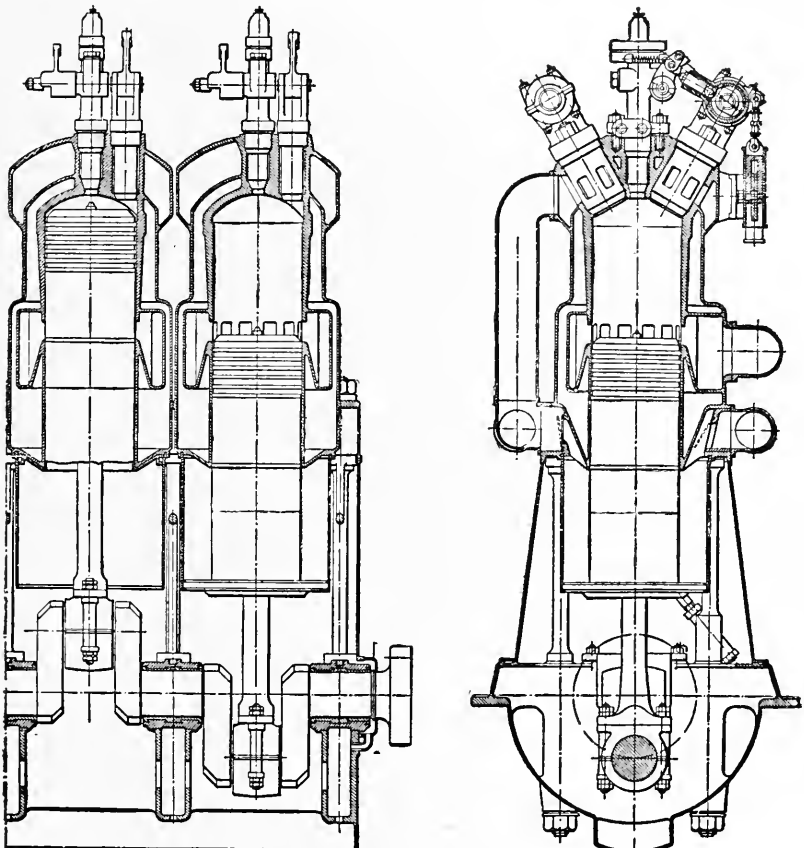


Fig. 322.—Two-cycle Single-acting Marine Engine for Heavy Oil—Diesel System. (Italian Design.)

acting. It will be seen, therefore, that with a piston speed of 800 feet a cylinder 30 inches in diameter can develop 933 I.H.P., so that a six-cylinder engine would produce 5,600 I.H.P. when running on the two-cycle system. The maximum temperature is over 1,000° F.

The Junker Engine (fig. 323) is an ingenious adaptation of the Wigzell engine (*v.* fig. 285) to obtain a virtual double-acting marine oil engine. It

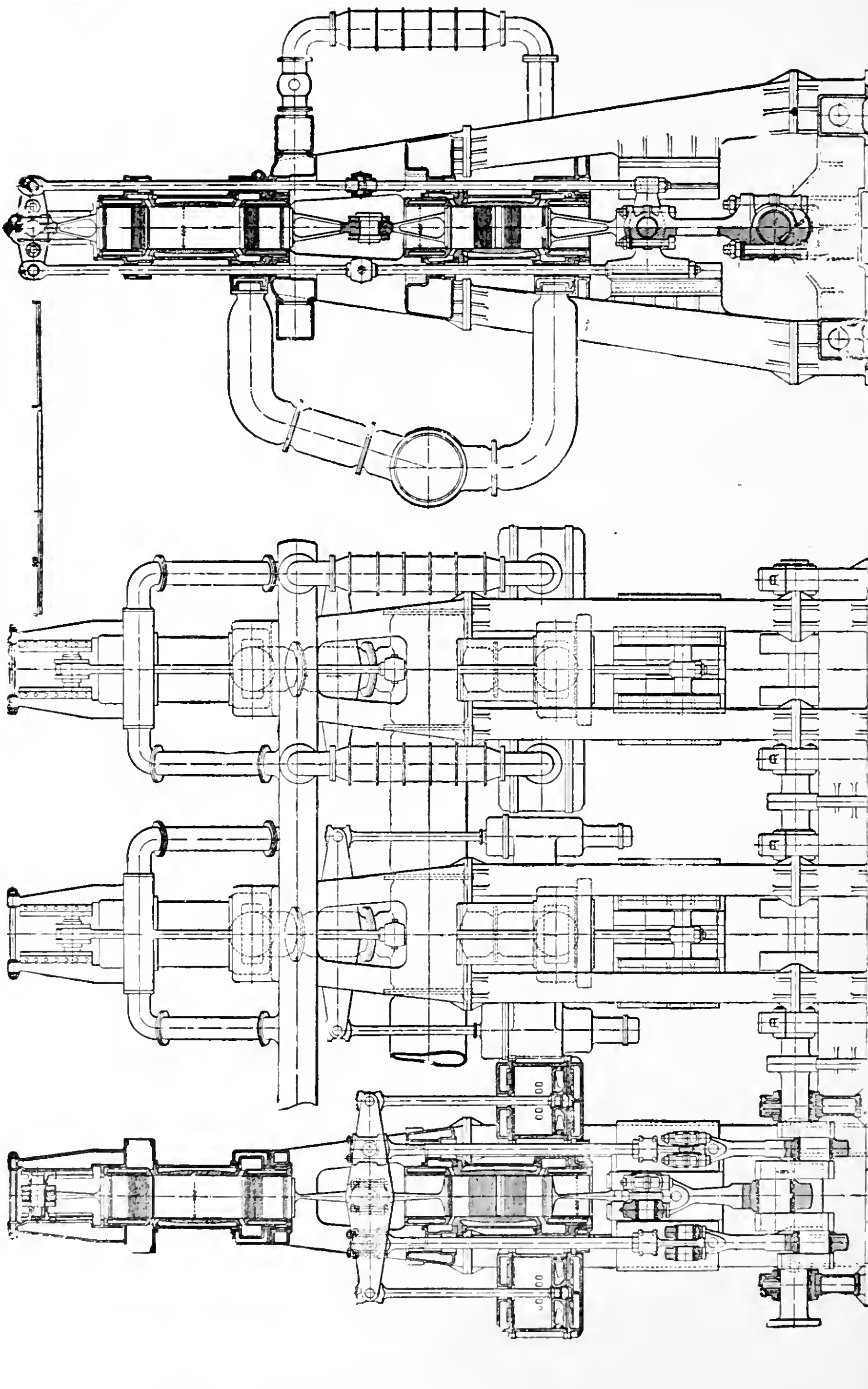


Fig. 323. Junker's Marine Heavy Oil Engine as fitted in a Hamburg-America Liner.

will be seen that to each engine there are two cylinders and three cranks; each cylinder has two pistons, which move in opposite directions, the one operating on the centre crank, the other on the outer pair of cranks, which are on each side of and opposite to it. Air is compressed between each pair of pistons as they approach each other; oil is sprayed in between them, and ignites in the way usual with the Diesel. At about seven-eighths the stroke one piston opens a port to exhaust, and very shortly after another port at the opposite end of the cylinder is opened by the other piston, to admit the blow through or scavenging charge of air, which is supplied by air pumps worked direct from the crosshead as shown; this blast clears the cylinder space between the pair of pistons of foul air, and leaves it filled with pure air to be compressed as before. In this case each cylinder is single-acting and on the two-cycle system, but there are two cylinders in line tandem operating alternately, so that the combustion is equivalent to one double-acting cylinder. There are three such engines to one shaft, and the air-compression cylinder is worked from the fore end of the crank-shaft in the common way. This engine, besides being ingenious, has some good points, such as no cylinder covers to give trouble at the joints; no air admission or exhaust valves to cause noise and wear and tear; the cylinders are ventilated, and so kept reasonably cool. On the other hand, the crank-shaft is an expensive one, and the number of side rods and connecting-rods large. This engine, like the Wigzell, will be perfectly balanced against vertical inertia forces, and nearly so for the horizontal ones, and so ought to run quietly, without vibration, and have an even torque; but also, like the Wigzell, it is exceedingly high, so that with a long stroke of piston it would be very inconvenient in most ships.

The Reversing of these Oil Engines is generally accomplished by using the compressed air in one or more of the oil cylinders to press on the pistons with the oil supply shut off. This requires storage of air in bottles, or the continuous charging of air vessels by the pumps; the latter is the obvious plan for large engines. The admission of the air can be regulated by hand, or the cam shaft can be shifted through an angle sufficiently great (about 30°) to time the delivery of both compressed air and fuel for the reverse motion of the engine. The starting and reversing in each case being effected by compressed air acting as the moving force on the oil cylinder pistons.

In Manœuvring with Oil Engines there is not the same elasticity observable that is possible with steam engines. With a steam engine the starting and reversing are quickly and more certainly done; and whereas it will run dead slow for hours with the cranks just passing the dead points, the ordinary oil engine cannot be trusted in the same way, for it is getting an impulse very occasionally, and should a sudden load be put on the propeller, as is the case often in a seaway, or by the action of the rudder in a twin-screw ship, there is great danger of the engine stopping at a critical moment and failing to start again promptly. Then there may be trouble with deposit in the cylinders and pistons when working under these conditions, so that in foggy weather great care will have to be exercised, for a temporary resort to a light and highly inflammable oil is hardly permissible on shipboard. No doubt, however, the methods now employed are sufficiently good for all practical purposes, even if not so convenient as those of the steam engine.

The Diesel Oil Engine requires better workmanship and material than is sufficiently good for the best of steam engines, inasmuch as the cylinder, etc., is subject to both high pressure, high temperature, and to a certain or rather uncertain amount of shock; there must be perfect accuracy in the fit of the valves and gear by which the fuel supply is controlled, and the amount of compression must be constant for steady and economic working; hence, while a steam engine may and often does perform its work with a fair degree of efficiency when in a state of disrepair, an oil engine cannot do so. A leaky piston and distributing valve is a source of loss of a sort in a steam engine, but it is one of danger to the oil engine. The consumption of fuel in a steamship does not materially increase in rate during a long voyage, say to Japan and back, but it is more than doubtful if the same will be said of the internal combustion engine driven ship. The difference in pressure per square inch on the sides of the pistons of a triple and quadruple engine will not exceed 120 lbs., while in the Diesel engine it is 500 to 600 lbs., and sometimes even higher; if, therefore, the pistons are not perfectly tight, the loss in the latter case will be much more severe than in the former, and flame escaping into the engine-room, a more serious thing than steam passing from one cylinder to another, or even to the condenser. For this reason the two-cycle oil engine is safer than the single-acting four-cycle engine, and any oil engine having cylinders open to the engine-room should be avoided, or when used should be kept in perfect repair.

The Double-acting Engine on the two-stroke cycle is much to be desired for large power, inasmuch as the cylinders may be fewer in number or less in diameter; this type of engine is, however, necessarily less efficient than the four-cycle single-acting one. Dr. Diesel stated that such an engine with three cylinders, developing 850 I.H.P., had been made at Nuremburg, and undergone its trials satisfactorily; and that there was in course of manufacture a similar engine with cylinders so large that over 2,000 I.H.P. is developed in each. This engine has cylinders $31\frac{1}{2}$ inches in diameter, with a piston-stroke of $41\frac{3}{4}$ inches, and it is intended to run it at 160 revolutions per minute.

Dr. Diesel recommended that engines of this kind should have six cylinders, "to ensure a regular turning moment and balancing," and added that "this number cannot be considered abnormally high; on the contrary, it must be accepted as the most suitable and proper number," and concluded that "the day of the large marine engine is already very near at hand." The s.s. "Selandia," of 2,500 I.H.P., had had her trials, and completed a maiden voyage fitted with the Diesel engine; this and subsequent voyages with her and other similar ships have proved the reliability of these engines in capable hands, and justified the optimism of the inventor and his friends.

It is interesting to note that these engines are capable of working satisfactorily with other than mineral oils and extracts; at a trial an engine was fed with nut oil fuel, and worked quite well in every way. The calorific value of this oil is 15,510 B.T.U., and the consumption of it was only 0·53 lb. per hour. The efficiency of this engine was, therefore, 31 per cent., and its thermal efficiency probably about 40 per cent.

It is possible to work a Diesel engine with any inflammable liquid which has a flash point under 450° F., and will consume completely, so that gaseous product only remains, which will wholly escape through the exhaust without leaving any solid or semi-solid deposit. Mineral oils, as pumped or dis-

TABLE CVIII.—Examples of Oil Engine driven Screw Ships.

Name of Ship.	When Built.	Principal Dimensions.	Tonnage.		No. of Screws.	Engines.		Cylinders of Each Engine.			Trial Results.			Consumption of Fuel per I.H.P.
			G.R.T.	Displacement.		N.H.P. Total.	Cycle.	No.	Diameter.	Stroke.	Revs. per Minute.	I.H.P.	Speed.	
		Length. Beam. Depth.	Tons.	Tons.					Inches.	Inches.			Knots.	Lbs.
Bandon, . .	1909	330 × 47·3 × 17·7	3,409	..	1	324	IV. S.A.	6	26·4	39·4	..	..	..	..
Selandia, .	1912	370 × 53 × 27·1	4,950	10,750	2	468	IV. S.A.	8	20·9	28·7	140	2,500	12·0	0·335
Annam, . .	1913	410 × 55·2 × 35·4	5,296	12,800	2	606	IV. S.A.	8	23·3	31·5	..	..	..	..
Wotan, . .	1913	405 × 52·5 × 26·8	5,703	11,700	1	550	II S.A.	6	23·3	43·3	..	2,300	..	..
Monte Penedo,	1914	351 × 50·1 × 23·7	3,693	7,500	2	340	II. S.A.	4	18·1	26·3	..	1,990	9·8	0·338
Fiona, . .	1914	395 × 53 × 38	5,219	12,560	2	835	IV. S.A	6	29·9	43·5	100	4,000	14·0	..
Riedeman, .	1914	525 × 66·4 × 33·3	9,800	19,920	2	906	II. S.A.	6	22·6	39·4	..	..	..	..
Panama, .	1915	410 × 55 × 35·4	5,239	12,890	2	568	IV. S.A.	6	24·8	37·8	128	3,375	12·67	0·3212
Hamlet, . .	1915	355 × 55·2 × 29·9	5,093	11,000	2	497	II. S.A.	3	23·6	35·4	..	..	..	..
Peru, . .	1916	425 × 55 × 27·5	5,570	13,300	2	568	IV. S.A.	6	24·8	37·8	139	3,685	12·75	0·3185
Glenamoy, .	1916	436 × 55·3 × 35·2	7,269	15,000	2	648	IV. S.A.	6	26·4	39·4	136	4,650	..	..
Clement Smith,	1917	293 × 47 × 28	3,250	..	2	227	II. S.A.	2	20·5	29·2	..	..	..	..

charged from the wells, are unsuitable, as they contain highly volatile constituents, which not only render them dangerous to store, but dangerous to use in the engine. Their flash point should not be under 200° F., and they should be free from water, sulphur, and asphaltum, otherwise they will give trouble. Asphaltum is thrown down in the cylinder as hard refractory coke, which cuts and scores the pistons, cylinders, and valves; the sulphur deposits tend to glue up the piston rings, valves, etc., so that when the engine has cooled down after working it is found set fast, and cannot be started without cleaning.*

The mineral oil residues, after the distillation of petrol, paraffin, and lubricating oils, are good for engine fuel; other crude oils are treated by exposure to the air, to get rid of the volatile parts, and reduce the sulphur, etc., to small proportions. It is, therefore, correct to speak of the Diesel as a *heavy oil* engine, rather than a *crude oil* one, as is often done.

The reversing is accomplished by means of a sliding cam shaft, on which are four cams to each cylinder for "ahead" going, and four for "astern" going. These cams operate the admission of air, fuel, compressed air, and exhaust. There is a lever by means of which the cam levers are lifted out of gear while the shaft is moved longitudinally, and another lever which shuts off the fuel and the highly compressed air for feeding it during the time the engine is being worked by compressed air at 300 lbs. pressure. The cam shaft is driven by special gearing from the crank-shaft, and can slide about 2 inches from "ahead" gear to "astern." It is said the reversing can be accomplished in 10 seconds by a skilled attendant. The air for this purpose is compressed to 300 lbs pressure by a two-stage pump driven by an independent 4-cylinder oil engine, which also is arranged to drive a dynamo for the purpose of generating electricity, whereby the other auxiliaries throughout the ship may be worked. There are two such auxiliary oil engines, each capable of developing 250 B.H.P. at 230 revs. per min. The air for delivering the fuel is taken from the reservoirs into which these auxiliaries deliver it, and further compressed to 750 lbs., or even more, by a small pump worked by a crank-pin at the fore end of each crank-shaft.

These engines are lubricated throughout by means of force pumps, and the pistons and rods are kept cool by the oil circulating through them.

The Westgarth-Carel engine, as was fitted in s.s. "Eveston," is shown in fig. 324. It was a four-cylinder four-crank arrangement on the single-acting two-stroke system, having the air pumps for supplying the air to scavenge and charge the cylinders at the back of the engine, and worked by means of levers in the usual marine fashion: the high-pressure air for manœuvring the engine and spraying the oil fuel is obtained from a quadruple Reavell pump worked from the fore end of the main crank-shaft. Fig. 326 is a section of one of its cylinders illustrating the methods adopted in this case for fitting the liners, the water jacketing and piston cooling, and the arrangement of the valves, all of which will be seen to be simple and efficient. The starting and reversing of these engines are effected by means of compressed air; the valves for this purpose are operated by means of levers and cams on an independent shaft driven by the main crank-shaft by the interposition of a vertical shaft and skew-wheel gearing, which permits of

* It is said that with a higher compression (70 atmospheres) these fuels can be completely consumed

change of *lead* by raising or lowering this vertical shaft through a few inches. Mr. Westgarth elected to work the deck and auxiliary machinery by steam generated in a small boiler by means of the waste heat from the exhaust

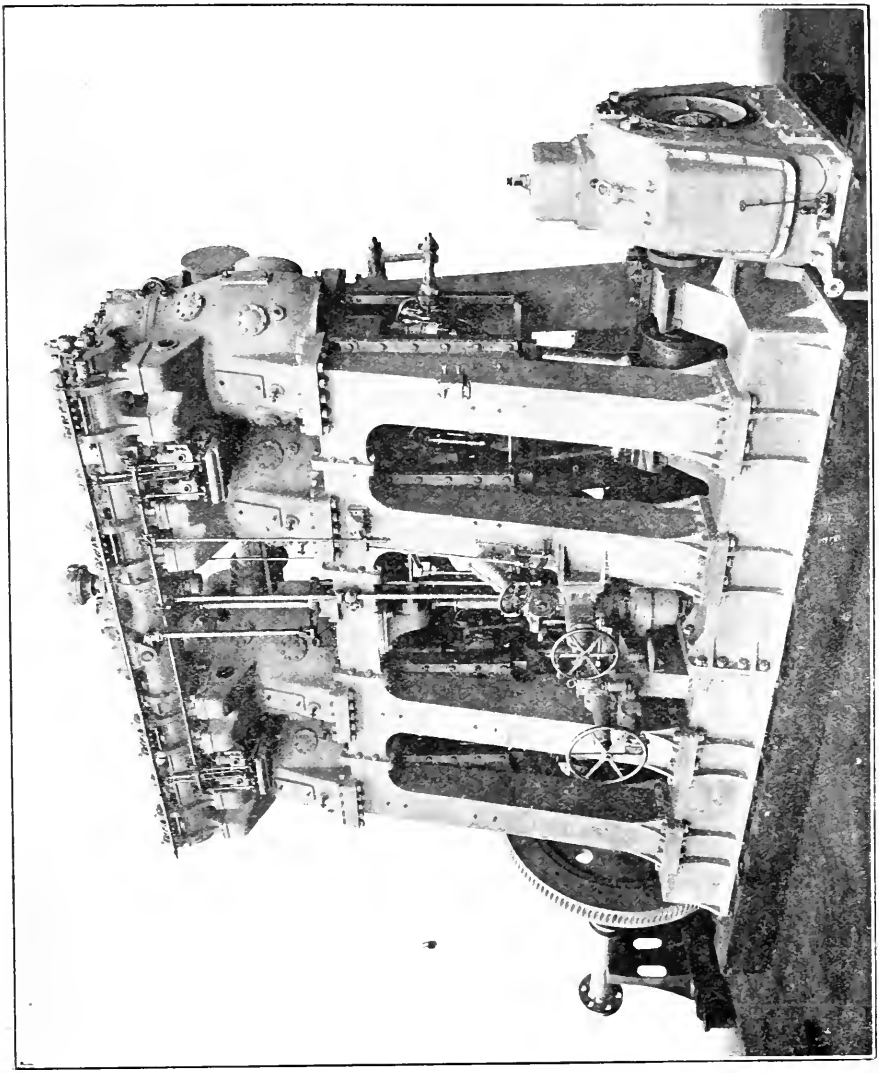


Fig. 324. Westgarth-Carel Oil Engine fitted in s.s. "Eveston."
Four cylinders 20 inches diameter × 36 inches stroke.

of the main engines, thereby getting over some practical difficulties with economy in working as well as prime cost.

Fig. 325 is the vertical transverse section of the four-stroke single-acting engines of the twin-screw ship, "Fiona," 5,219 tons and 4,000 I.H.P. They

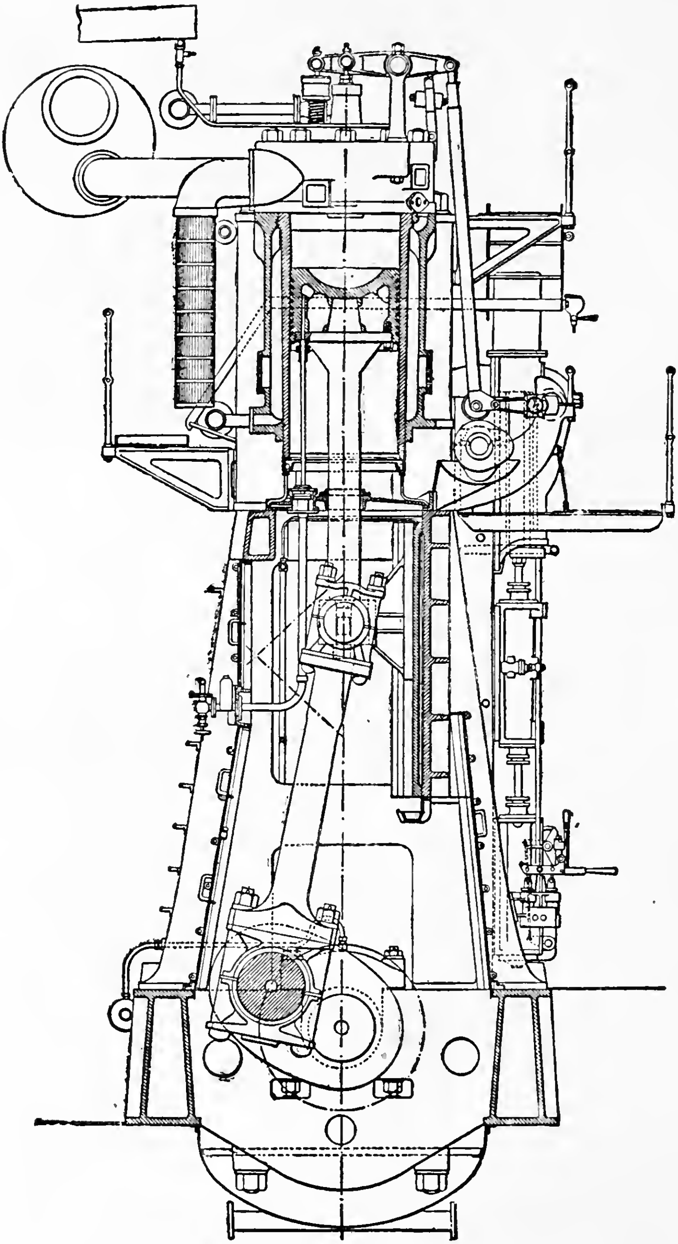


Fig. 325.—Sectional View of Main Engine of Twin s.s. "Fiona," 4,000 I.H.P., 835 N.H.P.

Each engine has 6 cylinders 4-cycle, 29·9 diameter × 43·5 stroke.

are the largest yet made for the mercantile marine, being collectively of 835 N.H.P., each engine having six cylinders, 29·9 inches diameter and 43·5 inches stroke of piston, and running at 140 revolutions per minute when at full speed. Their consumption of fuel oil is 0·32 lb. per I.H.P. hour of good quality.

Doxford's Double-piston Oil Engine is shown on fig. 328 as a single unit. It is on the two-cycle single-acting system in general principle like that of

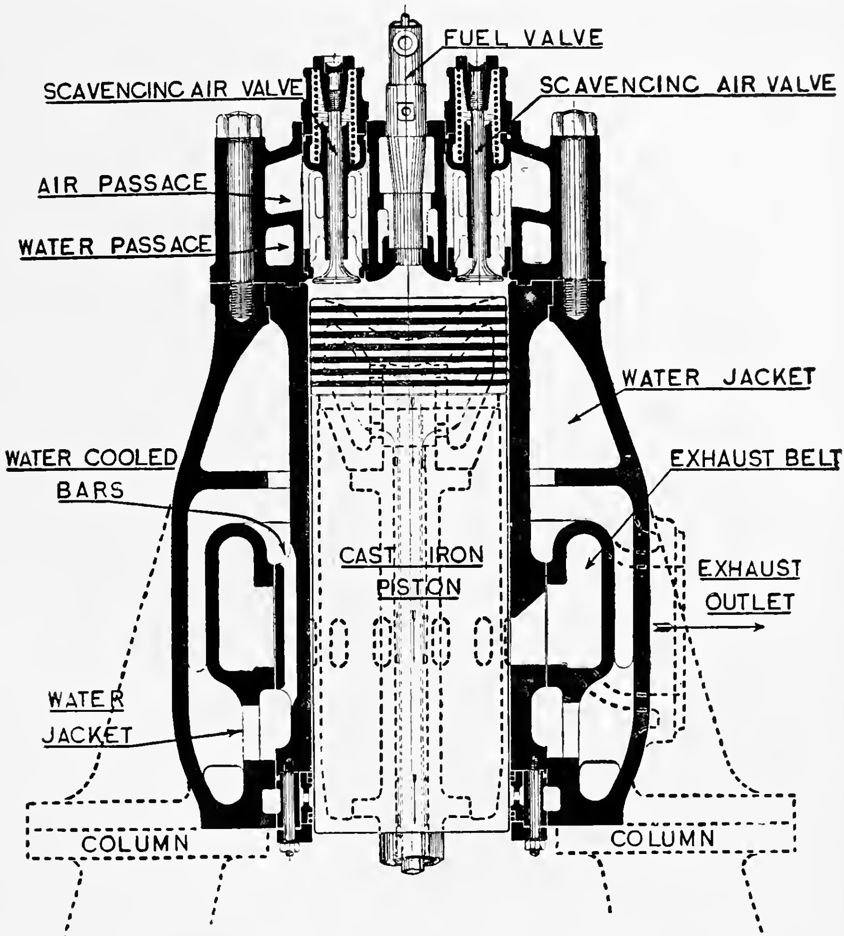


Fig. 326.—Section of Cylinder of Westgarth-Carel Oil Engine.
Diameter, 20 inches; stroke, 36 inches.

fig. 323, but with only one cylinder per unit. It is, therefore, without the double-acting effort of that engine, but nevertheless it possesses the other good features, such as working balance, without heavy stress on the columns and framing, and the absence of the "heads," which become so hot and dangerous. There is, however, a multiplicity of working parts per unit which are costly, heavy, and require attention, but in spite of these the engine is an attractive one, and should prove satisfactory on service

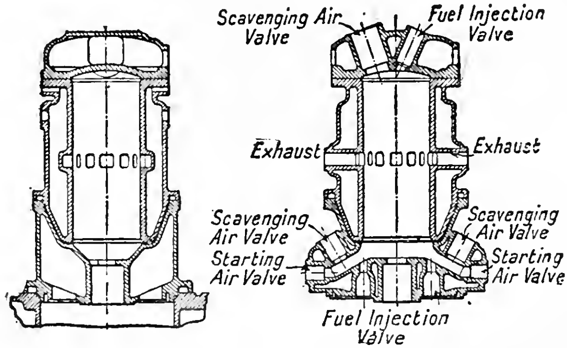


Fig. 327.—Cylinder and Heads of Double-acting Engine.

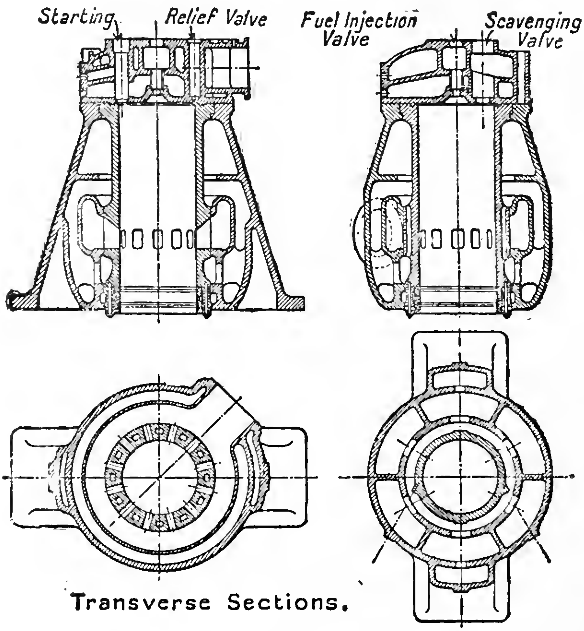


Fig. 327*a*.—Cylinder and Head of a Two-stroke Single-acting Engine.

The Semi-Diesel Engine has been in use now for some years; it has been well tried and proved to be specially adapted for using the safe high flash point oils and residuals in small power units where an ordinary Diesel engine

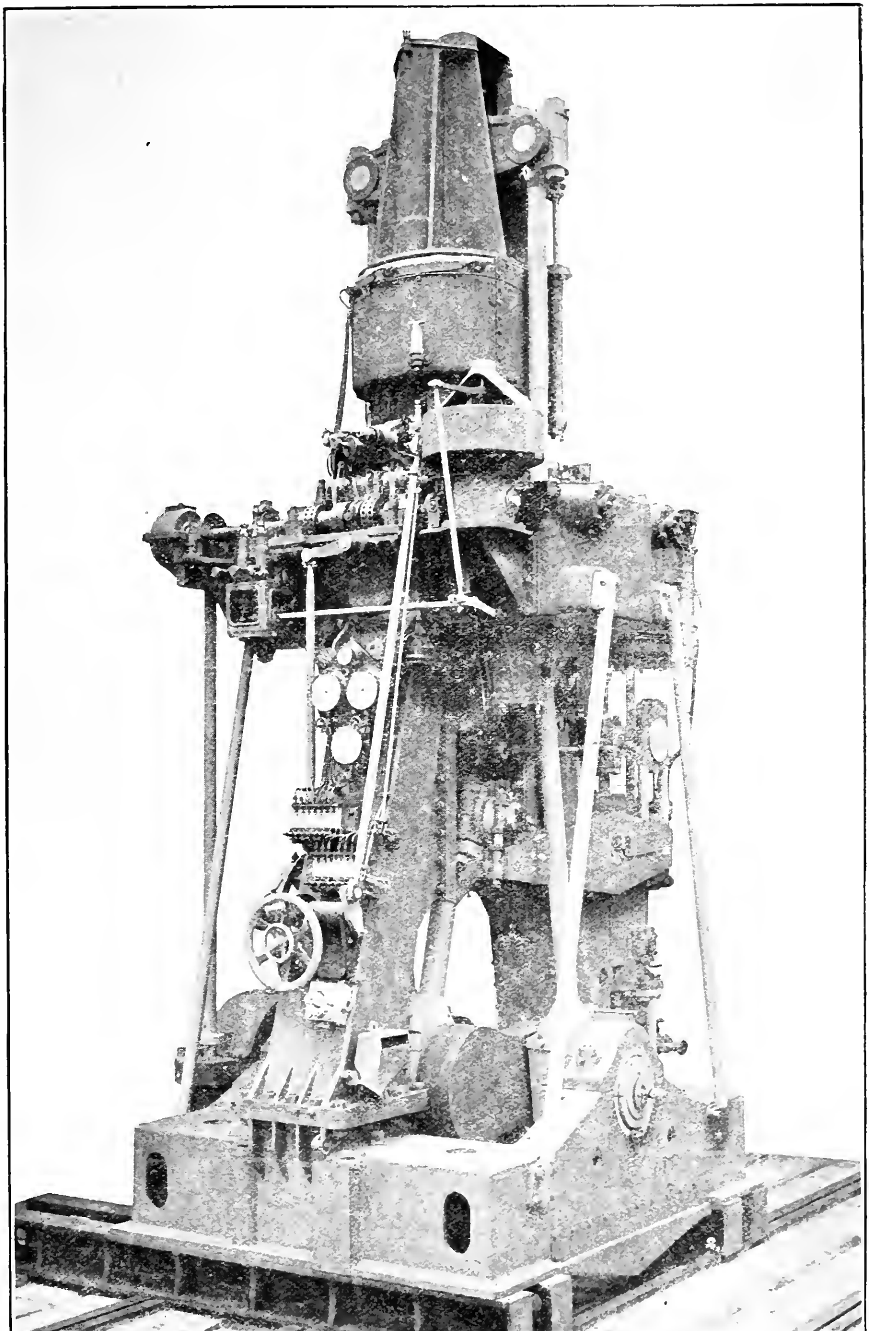


Fig. 328.—Doxford's Double-piston Three-crank Oil Engine Unit.

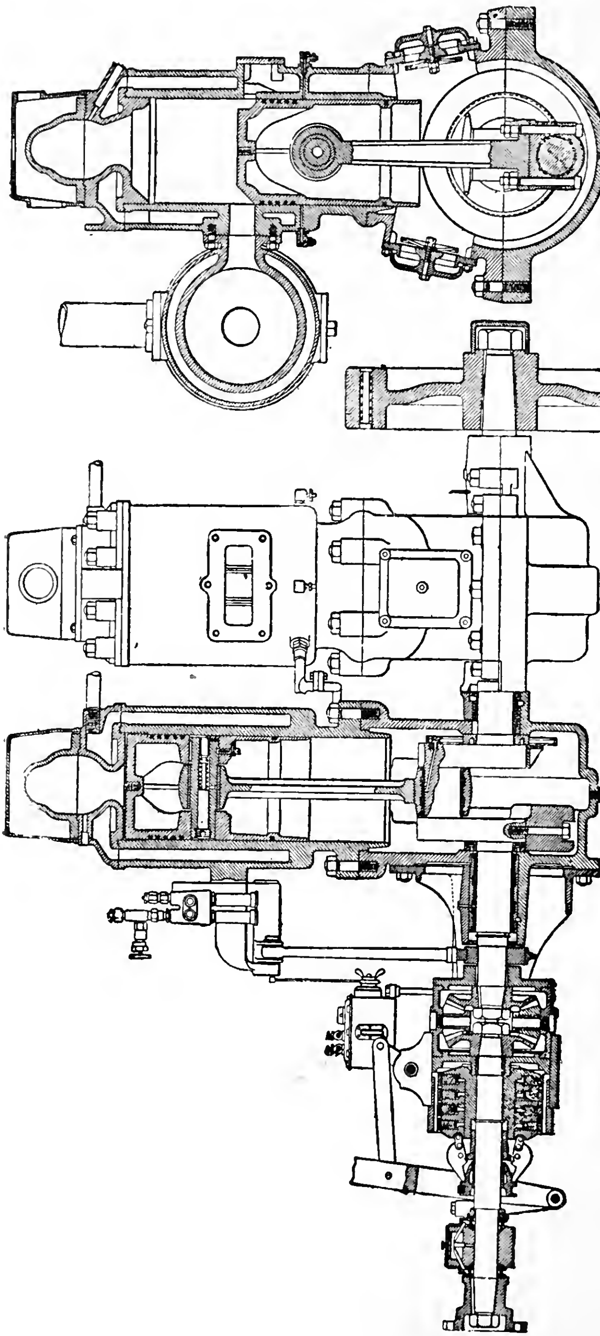


Fig. 328a.—British Kromhout Marine Oil Engines (Low-pressure Compression).

would be unsuitable as well as costly. It is now made by several well-known engine builders substantially, in all essentials, as shown by fig. 328. They may be on the four-stroke cycle, but are nearly always on the two-stroke system. The air for scavenging and recharging the cylinder every time is drawn in through valves to the air-tight casing of the engine itself, the lower side of the piston acting as the pump; the air thus stored is allowed to flow into the cylinder through a port which is opened on the final passage of the piston, after it has unclosed the exhaust ports; the air thus admitted is diverted upwards by the form of the piston so as to thoroughly drive out the products of combustion. It is, of course, obvious that only the bare amount of air is driven into the chamber, so that the charge is not very pure nor very dense, and, therefore, the combustion is not so efficient as that in the Diesel engine itself. It has, however, the merit of simplicity and consequent cheapness, inasmuch as the two-cycle operation is obtained without the use of separate air pumps, the compression is moderate, and the ratio of maximum to mean pressure comparatively small. On the other hand, the thermal efficiency is lower than that of the Diesel, and an igniter is necessary, but there is no need for such small piston clearance, and any variation in it due to wear of brasses, etc., is of no moment; the valves and gear are simple, and the fuel is injected with less pressure. Altogether, this engine possesses qualities which make it a cheap, useful, and economic one for the propulsion of comparatively small craft in the hands of an unskilled and partially trained staff. Moreover, it can be put in operation on shortest notice, being easily started with a little petrol or ether.

In Beardmore's engines the compression is limited to 150 lbs., so that after explosion it is only 300 lbs. The air for feed fuel is, therefore, only 400 to 500 lbs. Bolinders have made these engines extensively with units up to 16·5 inches diameter and 20 inches stroke. Beardmore's units are up to 14 inches diameter and 15 inches stroke. Plenty & Son have made units up to 13·2 inches diameter and 14 inches stroke.

In these engines the mean pressure is about 52 lbs., and the ratio of maximum to mean pressure about 5, practically the same as in the Diesel engine.

APPENDIX B.

Lloyd's Register Rules for Internal Combustion Engines for Marine Purposes (other than Diesel Type).

GENERAL.

Section 1.—In vessels propelled by internal combustion engines, the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

CONSTRUCTION.

Section 2.—1. The following points should be observed in connection with the design of the engines.

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed and easily accessible for examination and adjustment.

4. In engines of above 60 B.H.P. which are not reversible and which are manœuvred by clutch, a governor, or other arrangement must be fitted to prevent racing of the engine when declutched.

5. Efficient positive means of lubrication (preferably sight feed) must be fitted to each part requiring continuous lubrication.

6. If the engines are of the closed-in type, they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily drained metal-lined tray must be fitted to prevent leakage of either fuel oil or lubricating oil from saturating the woodwork.

7. Carburettors, where petrol is used, and vaporisers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburettor or vaporiser, a gauze covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic gauze diaphragms should be fitted either between the carburettor (or vaporiser) and cylinders, or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed; and the sparking coils must not be placed where they can be exposed to explosive vapours.

9. No exposed spark gap should be fitted.

10. In paraffin and heavy oil engines where lamps are used for ignition or for vaporising, these lamps should be fixed by some suitable bracket, and the flames enclosed when in use.

11. The circulating pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position,

and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. A bilge pump worked by the engines or an independent power-driven bilge pump is to be fitted, to draw from each part of the vessel. In open launches this bilge pump may be omitted provided suitable hand pumps are fitted.

13. The cylinders are to be tested by hydraulic pressure to twice the working pressure to which they will be subjected. The water jackets of the cylinders to 50 lbs. per square inch, and the exhaust pipes and silencer to 10 lbs. per square inch.

14. The exhaust pipes and silencer should be efficiently water cooled or lagged to prevent damage by heat, and if the exhaust is led overboard near the water-line, means must be arranged to prevent water from being syphoned back to the engine.

15. The machinery must be tried under full working conditions, the report stating the approximate speed of vessel, the number of revolutions of the engines at full power, both ahead and astern, and the lowest number of revolutions of the engines which can be maintained for manœuvring purposes.

Rules for determining sizes of shafts (*v.* Chap. xii.).

Fuel Tanks and Connections.

Section 4.—1. Separate fuel tanks are to be tested with all fittings, to a head of at least 15 feet of water. If pressure feed tanks are employed, they are to be tested to twice the working pressure which will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanised.

2. Strong and readily removable metallic gauze diaphragms should be fitted at all openings on petrol tanks.

3. Paraffin or heavy oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing petrol, should be provided with escape-valves discharging into pipes leading to the atmosphere above deck. The upper ends of all air pipes are to be turned down and pipes above 1 inch diameter are to be provided with gauze diaphragms at the end.

4. No glass gauges are to be fitted to fuel tanks containing either petrol, paraffin, or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges, or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible or they should be gauze-covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburettor or vaporiser. The fuel pipes should be led in positions where they are pro-

tected from mechanical injury, and can be exposed to view throughout their whole length.

8. The engine-room, and the compartment in which the fuel tanks are situated, are to be efficiently ventilated.

9. An approved fire extinguishing apparatus must be supplied.

Lloyd's Register Rules for Diesel Engines.

Nominal Horse-Power.—The following rule is to be used for determining the power of Diesel engines in regulating the fees for their survey, viz. :—

N.H.P. $= \dfrac{N \times D^2 \sqrt{S}}{80}$ in the case of single-acting engines of the 4-cycle type.

" $= \dfrac{N \times D^2 \sqrt{S}}{40}$ " " " 2-cycle "

" $= \dfrac{N \times D^2 \sqrt{S}}{20}$ " double-acting " 2-cycle "

Where D = diameter of cylinder in inches,
S = stroke of piston in inches in ordinary reciprocating engines,
= twice the stroke of piston in the case of engines of the "Junker" type,
N = number of cylinders.

Rules for the Construction and Survey of Diesel Engines and their Auxiliaries.

Section 1.—In vessels propelled by Diesel oil engines, the rules as regards machinery will be the same as those relating to steam engines, so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

construction.

Section 2.—1. In vessels built under Special Survey and fitted with Diesel engines, the engines must also be constructed under Special Survey.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lbs. per square inch, the diameters of the crank shaft of the main engines are not to be less than those given by the following formula :—

$$\text{Diameter of crank shaft} = \sqrt[3]{D^2 \times (A\,S + B\,L)},$$

where D = diameter of cylinder,
S = length of stroke,
L = span of bearings adjacent to a crank, measured from inner edge to inner edge.

The values of (A S + B L) are as given in the following table :—

TABLE I.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Values of the Coefficient.
4 or 6 cylinders.	2 or 3 cylinders.	·089 S + ·056 L
8 cylinders.	4 cylinders.	·099 S + ·054 L
10 or 12 cylinders.	5 or 6 cylinders.	·111 S + ·052 L
16 cylinders.	8 cylinders.	·131 S + ·050 L

For auxiliary engines of the Diesel type the diameters may be 5 per cent. less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should not be less than 1·33 times and the thickness not less than 0·56 times the diameter of the shaft as found above, or, if these proportions are departed from, the webs must be of equivalent strength.

9. Where no flywheel is fitted, the diameter of the intermediate shaft must not be less than given by the formula :—

$$\text{Diameter of intermediate shaft} = \text{coefficient } \sqrt[3]{D^2 \times S},$$

where D = diameter of cylinder,
S = stroke of piston,

and the value of the coefficient is given by the following table :—

TABLE II.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Value of the Coefficient.
4 cylinders.	2 cylinders.	·456
6, 8, 10, or 12 cylinders.	3, 4, 5, or 6 cylinders.	·436
16 cylinders.	8 cylinders.	·466

Where the stroke is not less than 1·2 times nor more than 1·6 times the diameter of the cylinder, (·735 D + ·273 S) may be taken instead of $\sqrt[3]{D^2 \times S}$.

10. In cases where flywheels are fitted, the following value of the coefficient may be taken for determining the size of the intermediate shaft abaft the flywheel shaft.

TABLE III.

4-Cycle Single-acting Engine.	2-Cycle Single-acting Engine.	Value of the Coefficient.
4 cylinders.	2 cylinders.	·405
6 ,,	3 ,,	·400
8 ,,	4 ,,	·409
10 ,,	5 ,,	·420
12 ,,	6 ,,	·427
16 ,,	8 ,,	·461

11. The diameter of the flywheel shaft must be at least equal to that of the crank shaft.

12. The diameter of the thrust shaft measured under the collars must be at least $\frac{21}{20}$ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

13. The diameter of the screw shaft must not be less than the diameter of the intermediate shaft (found as above) multiplied by $\left(\cdot63 + \frac{\cdot03\ \mathrm{P}}{\mathrm{T}}\right)$, but in no case must it be less than 1·07 T,

where P = the diameter of the propeller in inches,
T = the diameter of the intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be $\frac{21}{20}$ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

14. If the designed maximum pressure in the cylinders exceeds 500 lbs. per square inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{maximum pressure in lbs. per square inch}}{500}}$.

15. Where the cylinder liners are made of hard, close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $\frac{1}{15}$ of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

16. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lbs. per square inch, and must be perfectly tight at that pressure.

17. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

18. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent. above the designed maximum pressure in the cylinders and discharging where no damage can occur.

19. The air-compressors and their coolers are to be made so as to be easy of access for overhaul and adjustment.

20. In single-screw vessels, an auxiliary air-compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the manœuvring gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin-screw engines in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high-pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

AIR RECEIVERS.

Section 3.—1. Compressed-air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. *Quality of Material.*—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 4, par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent. less than that required with boiler plates.

5. *Tensile and Bend Tests* are to be made from the material of *each* receiver. When they are welded or riveted, the tests may be made, and the thickness verified, before the plates are bent into cylindrical form. In the case of seamless receivers, the thicknesses must be verified by the Surveyor before the ends are closed in, and at this time the Surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formulæ :—

Maximum working pressure in lbs. per square inch

$$= \frac{C \times S \times (T - 2)}{D} \text{ for thicknesses of } \tfrac{5}{8} \text{ inch and above.}$$

$$= \frac{C \times S \times (T - 1)}{D} \text{ for thicknesses below } \tfrac{5}{8} \text{ inch,}$$

where S = minimum tensile strength of the steel material used, in tons per square inch,

T = thickness of the material, in sixteenths of an inch,

D = internal diameter of cylinder, in inches,

C = coefficient as per table :—

Coefficient	77 for	seamless	receivers	of thickness	of $\frac{5}{8}$ inch and above,
,,	69 ,,	,,	,,	,,	below $\frac{5}{8}$ inch.
,,	54 ,,	welded	,,	,,	of $\frac{5}{8}$ inch and above.
,,	48 ,,	,,	,,	,,	below $\frac{5}{8}$ inch.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T = \frac{D}{17}\sqrt{P},$$

where T = thickness, in sixteenths of an inch,
D = internal diameter in inches,
P = working pressure, in lbs. per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates is to be tested by hydraulic pressure to twice the working pressure for pressures up to 200 lbs. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lbs. per square inch above the working pressure.

12. All receivers above 6 inches internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air compressor is fitted with a safety valve so arranged and adjusted that no greater pressure than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

PUMPING ARRANGEMENTS.

Section 4.—1. The requirements of the pumping arrangements for the various holds, double bottoms or other ballast tanks, etc., are to be the same as required in steam vessels of the same size.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled while the other is at work. In twin-screw vessels one bilge pump upon each engine will be approved. These pumps are to be arranged to draw from all compartments. Independent power-driven pumps may be fitted in lieu of these, if desired.

3. A steam pump, or equivalent power-driven pump, is also to be pro-

vided with connections to enable it to draw from all compartments and from the sea. It must be arranged to discharge overboard and also on deck to the fire service pipes. It must have at least one suction to the engine-room bilge distinct from those connected with the bilge pumps, so that it may be used for pumping from the engine-room when the bilge pumps are being used upon other parts of the vessel.

4. In addition to the above, where water ballast is used, the water-ballast pump must have one direct suction from the engine-room bilges. (This is in lieu of the bilge injection required with steam engines.)

GENERAL.

Section 5.—1. For the ordinary fuel tanks the requirements of Section 49 will apply. The daily service and other separate tanks must be tested, with all their fittings, with a head of water 12 feet above their highest points. They must be fitted with air pipes discharging above the upper deck. If they are fitted with glass gauges for indicating the quantity of oil contained in them, arrangements must be made for readily shutting off the gauges in the event of the breakage of the glass, and from preventing any damage from leakage of oil.

2. Special attention must be given to the ventilation of the engine-room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for cables for electric light.

4. It is recommended that all pipes conveying fuel oil should, as far as possible, be made of steel or iron, rather than copper, owing to the rapid corrosion of copper pipes when using oil containing sulphur.

SPARE GEAR.

Section 6.—The articles mentioned in the following list will be required to be carried, viz. :—

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, etc., fitted to it.

In addition, one complete set of valves, valve seats, springs, etc., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of the cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Diesel engines.

1 complete set of main skew wheels for one main engine.

2 connecting-rod, or piston-rod, top-end bolts and nuts, both for the main and the auxiliary Diesel engines.

2 connecting-rod bottom-end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.

1 set of coupling bolts for the crank shaft.

1 set of coupling bolts for the intermediate shaft.
1 complete set of piston rings for each piston of the main and of the auxiliary compressors.
1 half set of valves for the main and for the auxiliary compressors.
1 fuel pump complete for the main engine, or a complete set of all the working parts.
1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.
1 set of valves for the daily fuel supply pump.
1 set of valves for the water-circulating pumps.
1 set of valves for one bilge pump.
1 set of valves for the scavenge pump, where lift valves are used.
A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.
Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

APPENDIX C.

Bureau Veritas Rules for Internal Combustion Engines on Shipboard.

Propelling engines above 300 H.P. to be of reversible type. All engines to be provided with a friction brake and turning gear worked by hand. Above 1,000 H.P. the turning gear must be worked by a motor.

Cylinder covers must be fitted with safety-valves loaded to twice the maximum working pressure.

The pistons are to be cooled by efficient means.

In double-acting engines the piston-rod metallic packing is to be automatically adjustable.

Cylinder water-jackets are to be provided with a pressure gauge and drain cocks.

Each main engine to be provided with an air compressor capable of starting the engine 16 consecutive times without renewing supply of compressed air.

In twin-screw engines each air compressor should be of sufficient capacity to maintain the necessary pressure in the reservoir for working the two engines ; an auxiliary compressor must be provided sufficient to keep up the air supply in reservoirs when the engines are stopped, or when other compressors are not available ; it must, therefore, be driven by an independent motor.

Fuel oil pumps are to be strongly constructed, and easily overhauled when the engine is working.

A hand pump is to be provided for pumping fuel oil from the bunkers through filters to the gravitation tanks.

A governor is to be fitted to prevent racing of main engines.

Spare circulating pump, capable of passing 2·25 gallons per H.P. per hour, to be fitted independently of the others.

The tanks must have a steam-heating pipe near suction pipe, and be fitted with a safety-valve when worked under pressure. The suction pipes are to have duplicate independent filters, having triple perforated plates with gauze wire between them, and with covers easily removed for examination and cleansing.

All pipes for fuel oil and compressed air are to be of steel or solid-drawn copper with approved joints.

The cylinders are to be fitted with forced lubrication, and when of high revolutions the main bearings are to have forced lubrication.

$$\text{Diameter of crank-shaft} = C\sqrt[3]{D^2 \times S} \text{ inches.}$$

D is the diameter of cylinder and S the piston stroke, both in inches.

$\frac{S}{D}$.	C.	$\frac{S}{D}$.	C.
2·00	0·4930	1·40	0·5250
1·95	0·4975	1·35	0·5275
1·90	0·5000	1·30	0·5300
1·85	0·5025	1·25	0·5325
1·80	0·5050	1·20	0·5350
1·75	0·5075	1·15	0·5375
1·70	0·5100	1·10	0·5400
1·65	0·5125	1·05	0·5425
1·60	0·5150	1·00	0·5450
1·55	0·5175	0·95	0·5500
1·50	0·5200	0·90	0·5550
1·45	0·5225	0·85	0·5600

The above rules and values of C apply to four-cycle single-acting engines having 3, 4, 6, and 8 cylinders, or single-acting two-cycle ones with 4 or 6 cylinders, or double-acting two-cycle ones with 1, 2, or 3 cylinders.

The diameter of the intermediate shafts must be not less than—

$$0{\cdot}85 \times \text{diameter of crank-shaft} - 0{\cdot}2 = d.$$

$$\text{The diameter of thrust shaft at collars} = 1{\cdot}06 \times d.$$

$$\text{The diameter of propeller-shaft} = d_1 + \frac{\text{diameter of screw in inches}}{100}.$$

APPENDIX D.

TURBINES ON SHIPBOARD.

TURBINES are now employed very extensively for ship propulsion and, where high speed is necessary, almost to the exclusion of the reciprocator. So long as it drove the propeller shaft direct, its rate of revolution was very high for propeller efficiency, and too low for its own efficiency; consequently, its consumption per unit of power was much higher than obtained in land service. Further, while at full speed it compared quite favourably with the triple and quadruple reciprocator, at slower speed it failed to do so, and at cruising speed it was so much worse that special engines, either turbines or reciprocators, had to be fitted to Destroyers and Cruisers for use at those times when low power only was required for lengthy periods. Moreover, direct driving turbines were useless for slow ships of any kind, but especially were they out of the running in cargo ships whose speed is generally about 10 knots. It was, therefore, imperative, if the turbine was to be of general service on shipboard, that its speed must not be restricted to that of the propeller, and thereby be prevented from working with the minimum amount of steam, nor must the propeller be compelled to run at a rate of revolution inconsistent with efficiency on service and good handling of the ship in harbour, etc. The driver and the driven, therefore, had to be separated, so that each could move at its appropriate and most efficient rate of revolution. This has been effected, and the connection between them made in three different ways.

Sir Charles Parsons, as has been shown on p. 100, etc., introduced the spur wheel and pinion, formerly so common with the early screw ships, and not altogether unknown on a paddle ship. With helical teeth machine-cut and accurately designed and fitted, the objectiens formerly obtaining with wheel gearing have practically ceased, so that now even the most powerful turbines running at high rate of revolution (4,000 per minute) can be connected by double gear as shown in Figs. 329, 330, and 331, whereby the rate of the revolution of propeller is little more than would be the case with the reciprocating engines.

Experience has proved that the wear and tear on these helical steel teeth is very small indeed, and although there have been some broken teeth, it was attributable to the kind of steel first used being unsuitable, and to their faulty form at their roots; both these defects were removed, and the difficulties of the lubrication have also been surmounted.

The Ratio of Turbine to Propeller Revolution is necessarily great in cargo ships, inasmuch as the screw seldom exceeds 75 revolutions per minute, while the turbine may be run at 3,500 and even higher when of short lengths, as the Curtis generally is. The screw of the "Cairncross" on voyage makes only 62 revolutions; the ratio of the gearing is 26, so that the turbines make only 1,612 revolutions, which is quite too slow for modern practice. Fig. 329

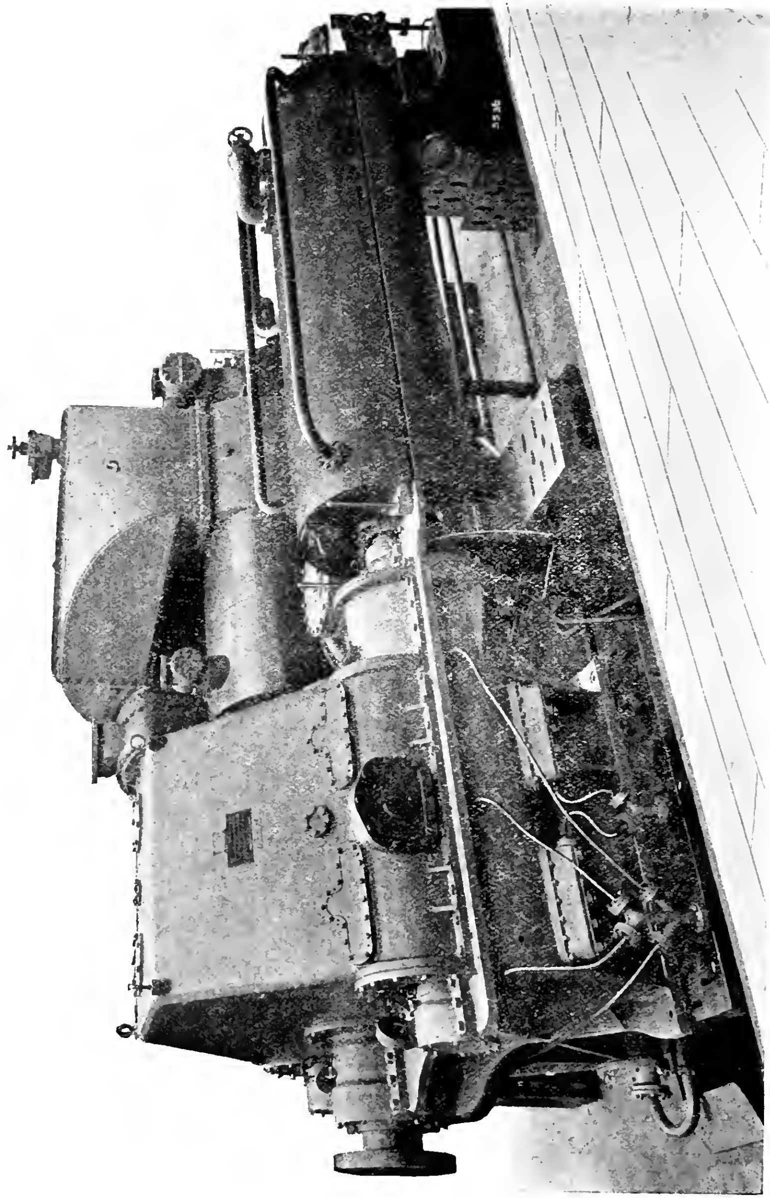


Fig. 329.—One Set of Turbines and Gearing to the Screw Shaft, R.M.S. "Tuscania" (Messrs. Alex. Stephen & Sons).

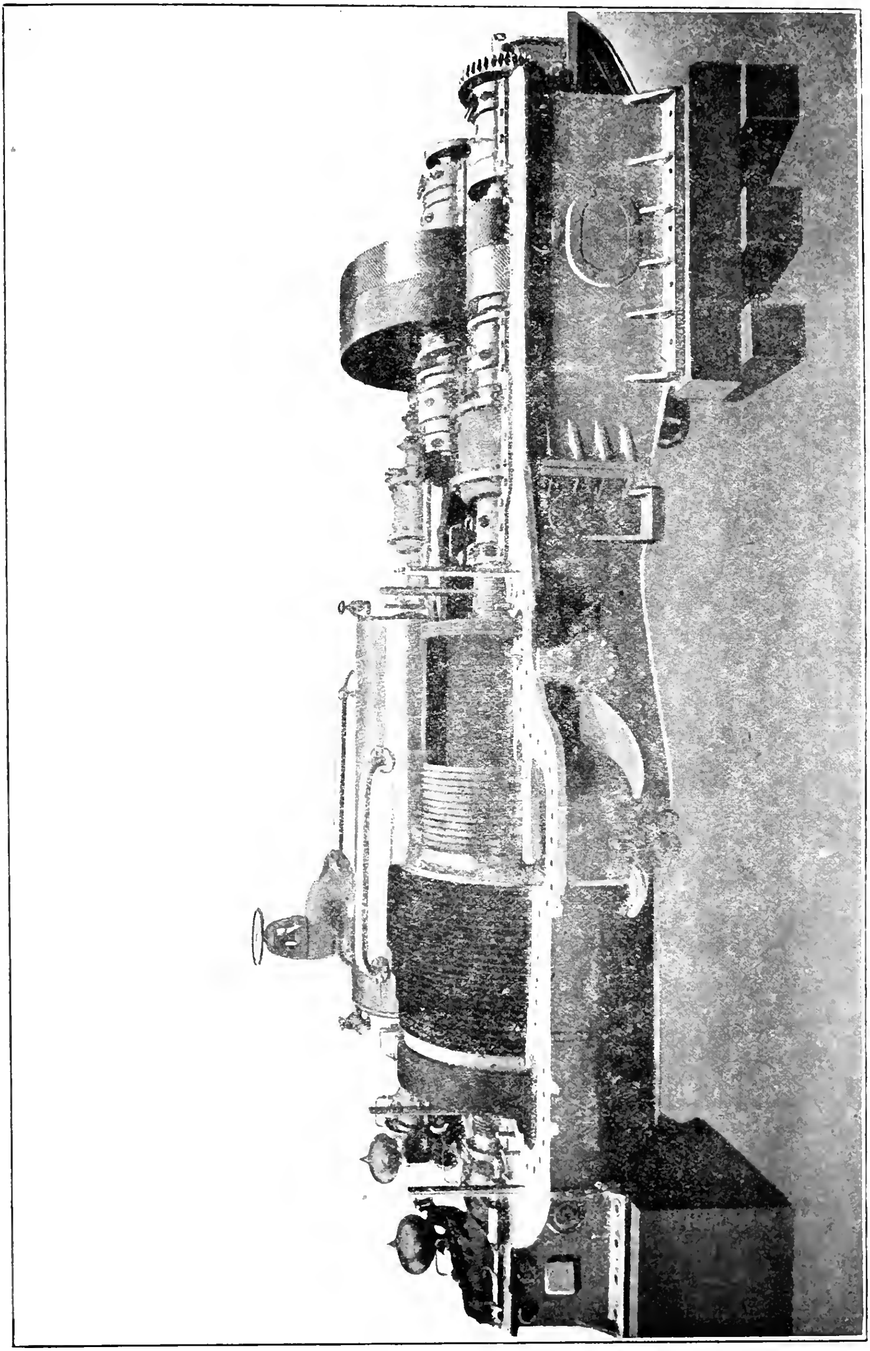


Fig. 330.—Geared Turbines of S.S. "Normannia," 1,864 Tons Displacement, 6,100 S.H.P., Speed 19·7 Knots.

is an example of what obtained on fast twin-screw express steamers with geared turbines. In this case the screws run at 310 revolutions per minute; the gearing ratio is quite small (6·4), inasmuch as the H.P. turbines ran only at 1,984, while the L.P. ran at 1,380. It will be seen that the pair of pinions are side by side in this case, and are not long like the "Vespasian's" (*vide* p. 100), nor had they the bearing interposed to support them, as is now generally necessary for good working. The frontispiece shows this arrangement in the R.M.S. "Tuscania," whose gearing is single with a ratio of 12·4—the screw running 137 revolutions at full speed, the efficiency being 97 per cent., with steam pressure 200 lbs., and vacuum 28·3 inches, the total S.H.P. is 10,900, and the consumption of steam only 11·25 lbs. per S.H.P. hour, and speed 17·65 knots.

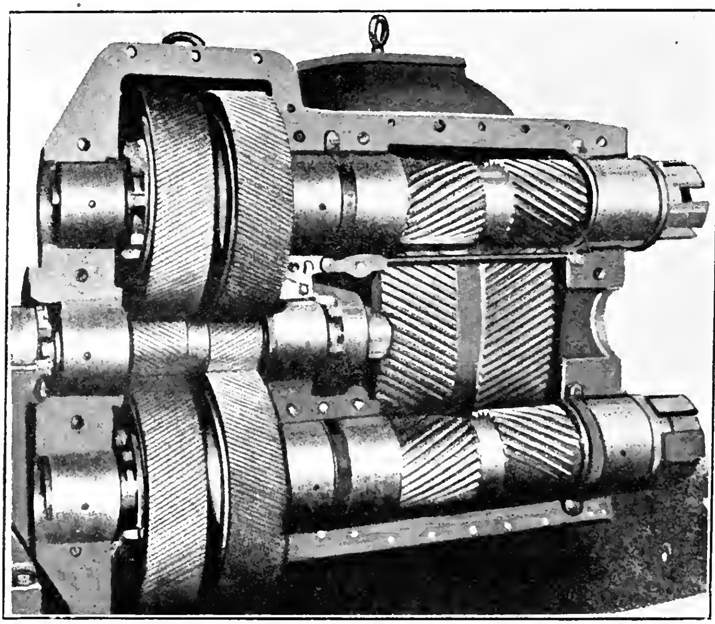


Fig. 331.—Double Reduction Gear of the Battleship "Nevada," U.S.A. Navy.

The spur wheels are 10 feet in diameter, and 60 inches long on teeth faces; the pinions being 10 inches diameter, their revolutions are 1,700.

To Avoid such Large Spur Wheels, to economise space, and generally to ensure good working, double gearing has been employed in the U.S. Navy with considerable success. It has now come into favour on this side of the Atlantic, and admits of employing turbines with a revolution as high as 4,000 per minute with screws at 81, the ratios being each 7, so that the combine ratio is 49. The main spur wheels and pinions for very large power are each as much as 36 inches long, with a good bearing between them.

Fig. 331 shows the ingenious and very compact arrangement applied on the U.S.A. battleship "Nevada" for coupling the cruising turbines to the screw shafts, the ratio in this case being 23·85.

Fig. 332 illustrates the MacAlpine arrangement of oscillating frames in which the pinions run, and permit of an automatic adjustment of each pair of pinions, so that they divide the work evenly between them. Considerable skill is also necessary in adjusting the teeth so that they bear evenly throughout their length *when running*.

The Helical Teeth of the wheel and pinions as originally made by Sir Charles Parsons were cut to an angle of 22·5°, with face line parallel to the axis; experience with these dictated a larger angle as being generally more satisfactory in working and less noisy. In fact, they are now practically noiseless when cut to 30°. The pinions are generally of nickel or chrome nickel steel, while the spur wheels have a band of ordinary forged mild steel as a tyre secured to the cast-steel or cast-iron centres. There should be no sharp angles at the roots of the teeth, their bases being well-rounded; in fact, the space between them may be semi-cylindrical.

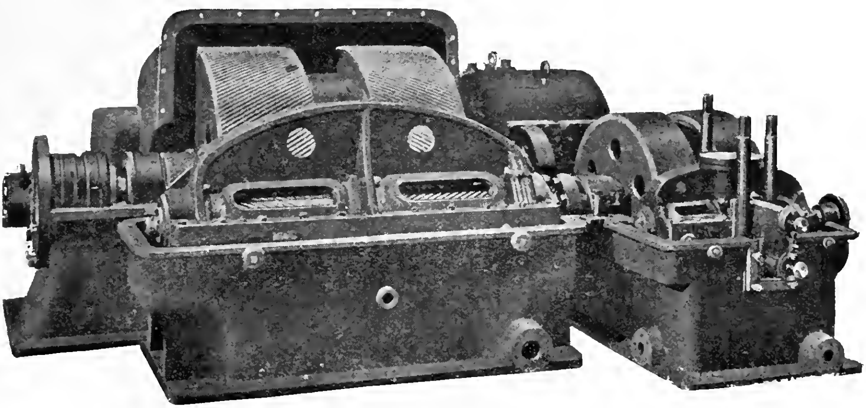


Fig. 332.—Double Gearing with Floating Frames as in U.S.A. Naval Ships.

The Turbine with Geared Connection can be run at any speed desired, and, consequently, there is no need for the very numerous stages the designer is compelled to put in the slow-working reaction turbine of the Parsons' type hitherto generally found on shipboard; it may be of the Curtis impulse variety with quite a few rows of blades, etc., and as economic of steam as those on central stations on shore. In the case of the "Normannia," the steam consumption per S.H.P. is only 12·0 lbs., as against 15·1 lbs. of her competing ship, the "Sarnia," with ordinary direct-coupled turbines on the same service. The large cargo steamer, "Cairncross," with geared turbines on a single screw shaft steamed side by side of the sister ship with triple compound engines, and used only 12·57 lbs. of steam per hour, as against the 15·18 lbs. with the triples. There is, of course, no question now as to the economic superiority of the geared turbines over simple ones or over quadruple engines so far as steam per H.P. is concerned. The mechanical efficiency is also quite satisfactory, as the gearing accounts for only 1·5 to 2 per cent.

of the gross power, and as that is S.H.P. taken beyond the gearing, the comparisons with the others are quite fair ones. The reversing has still to be done with a special "stern-going" turbine, as no one has yet ventured on trying a "gate" arrangement as in motor cars, or even the change gear as in small craft at sea with oil engines.

Electrical Driving of a Propeller is the second system of connection, and was suggested by many thoughtful minds in the early days of turbine propulsion, but the objections on the score of prime cost, space occupied, risk of danger to motors and generators, etc., sufficed to prevent experiment on a large scale. In the U.S.A., however, the advocates of this system were more fortunate, inasmuch as the naval authorities were persuaded to fit their new collier, "Jupiter," with such an installation.

The "**Jupiter**" is of 20,000 tons displacement, and has attained a speed of 15 knots by means of turbines whose S.H.P. is 6,940. The propeller runs at 116 revolutions per minute, and the steam consumption is only 12 lbs. per hour, against 14 lbs. of a sister ship with reciprocators. It is, however, claimed for the "Ljungstrom" turbine that with it and a similar electrical drive the consumption will be only 7·4 lbs. of steam per hour on the main engines only, as against 10·1 for double-geared and 9·0 for hydraulic drives, all of them using superheated steam; the Ljungstrom by 260° added and the hydraulic by 200° F. These are compared with a quadruple engine using 12·71 lbs. with no superheat, and an ordinary turbine drive with no superheat of 11·86 lbs.

With the Ljungstrom Turbine there are, as shown on p. 140, Fig. 58, two rotors having their blades interlaced and moving in opposite directions; each drives an alternating dynamo, which produces a three-phase current of about 50 alternations per second at 800 volts. The alternators of each are electrically locked so as to ensure an exactly equal speed and power with each half of the turbine. It is intended that there shall be two such sets to avoid dangers from breakdowns. The two alternators in each set will work in parallel. There is only one pair of poles to each of them; they supply current to two motors having five pairs of poles, which consequently revolve at a fifth the speed of the turbine. The motors have pinions with helical teeth, which drive a spur wheel on the screw shaft. In this way the ratio of revolution of turbine to propeller is nearly 50, so that a high rate obtains with the turbine (3,500 revs.). The screw can be reversed by means of the motors, so that there is no need of an astern-going turbine or reciprocator for this purpose. But again, the installation is not so elastic in operation as is the steam one with reciprocators, as slowing down requires additions in the way of resistances, etc. The same remarks and arrangements more or less prevail with all the electrically driven schemes; moreover, with the short Curtis turbine, 3,500 or even a higher rate of revolution is possible. A ship of considerable size is being fitted with such an installation as the above of 5,400 S.H.P. with twin screws.

In the case of the Ljungstrom turbine the electrical is certainly the most convenient for coupling it to a screw shaft; but it can, and perhaps will, be effected generally by wheel gearing, inasmuch as by the interposition of a hunting wheel between one pinion and the spur wheel of the first pair of gears, their original reverse motion is converted into combined action on the spur wheel shaft, which carries the second pinion geared into the spur wheel on the screw shaft.

The Ljungstrom turbine is a very interesting instrument, and has been most carefully designed and developed, so that the objections to it from a practical point of view have been fairly met and most of them ingeniously got over ; time, however, alone will permit of coming to definite conclusions as to its place in marine engineering. The efficiency of the system is fairly high, and from experiments made in Sweden with sister ships the consumption was 35 per cent. in its favour, as against the direct turbine drive, being only 0·89 lb. of coal per S.H.P., while the consumption in another ship with triple expansion engines was 1·09 lbs., when added superheat was 280° F. and Howden's forced draught.

Hydraulic Transmission is the third system. It was introduced by Dr. Föttinger, and extensively used in Germany. Fig. 332 shows the impeller and motor of the centrifugal pumps, and Fig. 333*a* the arrangement of the system in a ship. Dr. Föttinger claims that while mechanically it is not so efficient as either the wheel gearing or the electrical transmission, it is practically nearly as economic, inasmuch as he is able to supply warm feed water to the boilers ; and for prolonged astern movement, it will be much more economic and safer than the reversed turbine. The efficiency, however, when admitting the heat gain is at best 92 per cent. in the large installations and less in smaller ones.

The Modern Turbine on Shipboard is either altogether of the impulse order or is a reaction with an impulse leading portion. In this country the Parsons and the Curtis are those most generally used ; there are a considerable number of engine builders who construct turbines, and each has a design with some distinctive features of its own. The Ljungstrom is gaining favour with those who set economy of steam consumption as the criterion, but its employment with copper ingot (g.m.b.) at £130 per ton, the electrical method for the present is somewhat handicapped.

In the U.S.A. the Curtis is now a favourite turbine, but the improved Parsons has still a good following. The Zoelly, Rateau, and some others have also their supporters.

France has stuck to the Rateau as a rule, and as it is a good turbine, they do well to support their very able countryman and distinguished engineer.

Germany before the war used the Parsons turbine freely—the Zoelly, A.E.G., and some other forms of the Parsons or Curtis in Teuton guise were, however, coming into general use. But Germany, like the U.S.A., stuck to the reciprocators for warships much longer than the British did ; in fact, at one time the U.S.A. naval authorities reverted to the reciprocator, but that was largely due to their adopting methods and designs which were less suitable to naval conditions than our designs were.

Italy has employed the Parsons very generally, as indeed other countries have naturally followed the lead of Great Britain.

Japan has generally adopted the Curtis for naval purposes.

The Weight of a Turbine Installation complete, direct-driven and with cylindrical boilers, is about 300 lbs. per S.H.P., as against about 400 lbs. of a triple compound engine one, both in ocean-going ships of fairly high speed.

In cross-channel express steamers a direct-driven turbine installation will weigh about 200 lbs. per S.H.P. with cylindrical boilers, and about

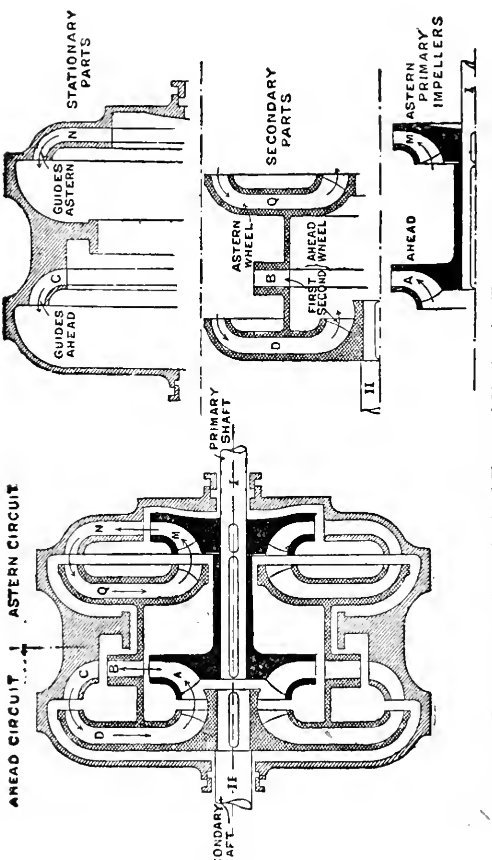


Fig. 333.—Diagram of the Principal Type of Hydraulic Transformer (Föttinger).

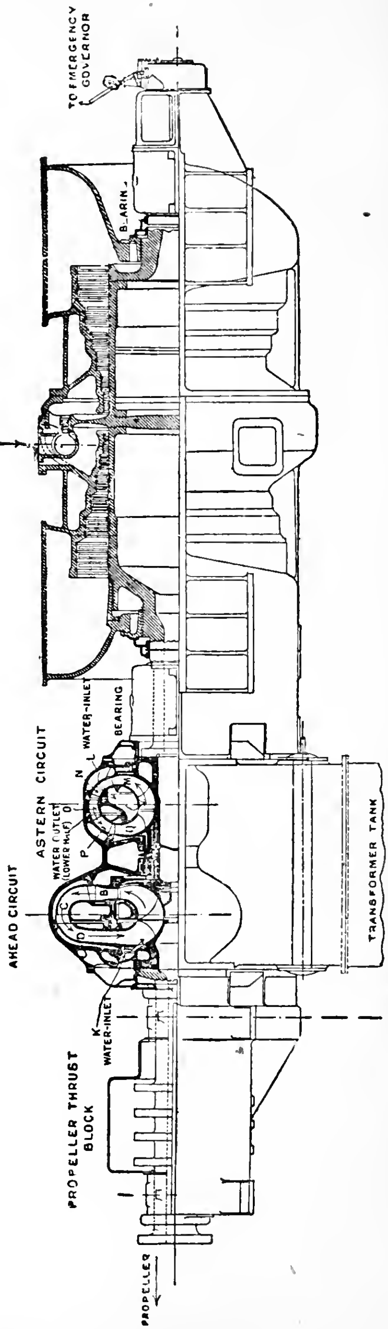


Fig. 333a.—Föttinger's Turbine Transformer (about) 20,000 H.P., 1,700/320 revolutions per minute.

160 lbs. with water-tube boilers. A geared-turbine set with water-tube boilers will be as low as 140 lbs. per S.H.P., with propeller revolutions 270 per minute.

Mr. A. Cleghorn * states that if the fuel consumption of a direct-coupled turbine installation using saturated steam is taken as 100, that of single-geared installations will be—in express ocean liners 89, in ocean mixed passenger and cargo steamers 93, in cross-channel expresses 91·5, and in ordinary cargo ships 83·5. With single gearing and steam superheated 100° F., the relative consumption is 84·5, 88, 86·5, and 79. With double gearing and 200° F. superheat the figures are 75, 74, 73·5 and 66 respectively.

The weight of such installations as the above, taking the direct-coupled turbines using saturated steam as 100, he gives the figures for the express ocean steamers as 93, 93·5, and 88·5; for mixed passenger and cargo ships, 80, 81, and 72; for the cross-channel expresses, 75, 76, and 68; and for the tramp steamers, 79, 79·5, and 72·5. Mr. Cleghorn gives as actual examples the following:—

Actual Steam Consumption on Two Large Liners.

Description of Machinery.	S. H. P.	Steam at Turbines.	Consumption in Lbs. per Hour.		
			Main Turbines.	Auxiliaries and other Purposes.	Total for all Purposes.
4 Shafts Direct-coupled Turbines,	23,000	Saturated, .	11·2	2·2	13·4
2 Shafts Geared Compound Turbines, 1 H.P. and 1 L.P. to each, ...	14,500	85° F. Super-heat,	9·7	2·3	12·0

TABLE CIX.—Methods of Transmission from Turbine to Screw Compared. (Cleghorn.)

Points of Comparison in a Ship of 20,000 S. H. P.	Method of Transmission.		
	Double Wheel Gearing.	Hydraulic Transformer.	Geared Electrical Generator, and Motor Driven.
Revolution of turbines per min.,	2,400 and 1,500	1,000 (about)	2,400 and 1,500
,, screws ,,	90	140	90
Transmission efficiency, . .	0·965	0·92	0·885
Relative fuel consumption, .	100	123	109
,, machinery weights, .	100	109	108

The Lubrication of the Gearing is very important, seeing how much of the energy transmitted through it is retained and transformed into heat, which, if not removed in some way, might intensify and become dangerous.

* Presidential Address to the Institution of Engineers and Shipbuilders in Scotland

Primarily the oiling of the surfaces of the teeth is to minimise the friction due to contact rubbing, but since in spite of it some $1\frac{1}{2}$ to 2 per cent. of the power fails to be transmitted, it must reappear as heat, so that for each horse-power there will be 0·848 B.T.U. per minute or 50·88 per hour. If the feed water per S.H.P. is 12 lbs., this heat would raise its temperature by 4·24° F.; therefore not so valuable an asset as Dr. Föttinger claims for his system. It is sufficient, however, to demand some method of cooling the lubricating oil beyond atmospheric contact. If the lubricant may be raised by, say, 60° F., and its specific heat is 0·35, it will require $3\frac{1}{2}$ lbs. of water for 10 lbs. of oil passed; also that, per horse-power-hour, 0·848 lb. of water will be necessary to keep the oil cool. Under these circumstances engines of 10,000 S.H.P. will require circulating water to the amount of 8,480 lbs. or 848 gallons per hour. It is evident, therefore, that in practice a mere oil bath would fail, hence the squirting of cold oil along the angle made by the wheel and pinion faces, and allowing it to be thrown off into a chamber at the bottom of which it runs into a cooling receiver, from which it is pumped into a tank overhead, and so keeps up the supply, is found quite successful and satisfactory as well as economic of oil. In the tropics the cooling water for the oil will, of course, increase in quantity, as does that of the condenser, so that if sea water is at 80° F. and the limit of heat for satisfactory lubrication is 110° F., then the cooling water will be double that named above, and so ships which pass through or trade in tropic seas must have this provision. It need hardly be said, however, that the viscosity of oil at 110° is very low, and, therefore, so high a temperature is to be avoided if possible for continuous running. The extra quantity of cooling water will be cheaply bought by the saving in power of transmission effected by the better viscosity of the oil.

APPENDIX E.

SUPERHEATED STEAM AND SUPERHEATERS.

Before the compound engine was generally adopted for ship propulsion superheaters were in general use to dry the steam and fortify it with heat sufficient to carry it dry at least to the inside of the cylinders. The limit of temperature was then 400° F., so that the amount added was about 125° F. with box boilers and with cylindrical 100° at most; usually, however, it was not much more than half these, as the reheating was effected then by means of the waste heat in the uptakes, which was not supposed to be much over 400° F. On the advent of the compound engine superheaters were not so popular as formerly as they had become somewhat troublesome and a cause of interference by the Board of Trade officers. Moreover, the gain in economy by the adoption of compound engines was more than that obtained by superheating in an ordinary engine, and satisfied the shipowner as relieving him of a troublesome appendage. Altogether, superheating dropped out of the marine engineering practice, and was only revived again after experience with it on land installations had demonstrated the possibilities to be great, and the new means practically free from the old objectionable features since metallic packing for rods, and mineral oils for internal lubrication were used. To-day steam at 200 lbs. pressure is in general use; its natural heat as saturated is 387° F., or about the limit of the superheated of former days. It is fairly dry in the steam space of the boiler as it comes from the water, but any fall in temperature, either from fall in pressure or in loss on transmission by radiation, conduction, etc., it will become foggy and deposit moisture on the surfaces it comes in contact with, even if they are no cooler than itself. If these surfaces are cooler, as are the walls of a cylinder, deposition will take place and its efficiency, as also that of the engine itself, will suffer. If, on the other hand, it is fortified against such contingencies by possessing a reserve of heat, no such ill effects will follow.

Steam is a bad conductor of heat, and consequently when in a stagnant state will absorb no further heat; it must be in motion over hot surfaces if it is to become superheated; hence its passage through a large number of small tubes is the most effective way of superheating it, inasmuch as it all must then be near a hot surface and pass over it rapidly; moreover, none of it can possibly be stagnant, as was the case in the old box superheaters of fifty years ago.

The specific heat of steam at 380° F. is 0·574, increasing with its temperature rise till at 600° F. it is 0·6572.

The Amount of Heat required to superheat a pound of steam from its natural temperature T to a temperature T_s can be calculated sufficiently near by assuming the specific heat to be a mean of the two values. For example, if it is required to raise from 200 lbs. with a natural temperature of 380° to 600° F. the specific heat may be taken as $\frac{0{\cdot}574 + 0{\cdot}6572}{2}$, or 0·6156.

Then amount of heat is = mean specific heat × (T_s − T).

In the above case it will be 0·6156 (600 − 380) = 135·4 B.T.U. for each pound of steam.

The Total Heat of Superheated Steam, calculated by Prof. Peabody's formula, is as follows :—

$$\text{(i.)}\quad \text{Total heat} = 0{\cdot}4805\,(T_a - 10{\cdot}38\,\sqrt[4]{P}) + 857.$$

T_a is the absolute temperature Fahr. ; P is the absolute pressure.

Prof. Ripper's formula is

$$\text{(ii.)}\quad \text{Total heat} = H + 0{\cdot}48\,(t_s - t_1),$$

where H is the total heat of saturated steam at a pressure p and temperature t_1 and t_s that to which it is to be raised.

Example.—Steam at 200 lbs. gauge pressure is to be raised to a temperature of 500° F.

(*a*) By Peabody's formula,

$$\begin{aligned}\text{Total heat} &= 0{\cdot}4805\,\{(800 + 461) - 10{\cdot}38\,\sqrt[4]{200 + 15}\} + 857 \\ &= 442{\cdot}7 + 857 = 1{,}299{\cdot}7^\circ\ \text{F.}\end{aligned}$$

(*b*) By Ripper's formula,

$$\begin{aligned}\text{Total heat} &= 1{,}200 + 0{\cdot}48\,(500 - 387{\cdot}5) \\ &= 1{,}254.\end{aligned}$$

Superheated Steam transmitted through pipes is greater than that of saturated steam. It may be calculated by the following formula, where d is the diameter in inches, L the length in feet ; W the weight of a cubic foot of the steam, whose pressure is P_1 at entry and P_2 at exit.

$$\text{Weight per minute} = 87\sqrt{\frac{W\,(P_1 - P_2)\,d^5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}}.$$

The Maximum Work possible from a pound of steam, whose pressure is 200 lbs. and temperature 600° F., is 453 B.T.U., while at 500° it is 405 B.T.U., as against 340 as saturated steam. This theoretical gain cannot, of course be maintained in practice, but even then it is very material.

The Economy in using Superheat in practice is proved by the following figures obtained as the result of carefully made experiments in test houses, as the net consumption of the engines, etc., only.

TABLE CX.—Steam consumed in lbs. per S.H.P.-Hour, Boiler Pressure 180.

S.H.P.	Triple Expansion Reciprocator.		Ordinary Turbine. Vac. 28·5.		Combined Reciprocator and Turbine.	
	Vac. 26. Saturated Steam.	Vac. 26. Superheat to 572° F.	Vac. 28·5. Saturated Steam.	Vac. 28·5. Superheat to 572° F.	Vac. 28·5. Saturated Steam.	Vac. 28·5. Superheat to 572° F.
500	16·50	11·75	14·00	11·75	18 to 22	13 to 15
1,000	15·25	11·25	11·75	9·75	12 to 13	9·5 to 9·9
1,500	14·75	11·00	11·25	9·50	11·25	9·0
2,000	14·50	11·00	11·00	9·25	11·00	8·7

The Heating Surface of a Superheater per pound of steam heated depends largely on the difference in temperature between the hot gases available and the steam as delivered. If it is considerable, as much as 6 B.T.U. per square foot can be transmitted for each degree of difference Fahr., when both surfaces are clean, so that if steam of 200 lbs. pressure is to be raised to 500° with hot gases of 600° temperature, each square foot should transmit 6 (600 — 500) or 600 B.T.U. per hour.

Each pound of such steam requires (500 — 387) × 0·599, or 67·7 B.T.U. If then the consumption is 12 lbs. of steam per H.P.-hour, the surface will be 67·7 × 12 ÷ 600, or 1·354 square feet per horse-power.

It will be seen that if high superheat is required the heating gas temperature must be much higher or the surface of the heater larger.

In ordinary practice the temperature in the uptake should not exceed 600° F., and when at a speed of ship lower than given by the full effort of the boilers the temperature will be lower and may be only the old 400° F. of former days. With the Howden air heating for the furnaces the 600° F. may not be wasteful, and even a higher temperature for the sake of superheating be warranted, but to ensure such it is better not to trust to the smoke-box contents, but seek a source of heat elsewhere. For this reason the modern superheater is inserted in the boiler tubes themselves of a tank boiler, and in the tube chamber of a water-tube boiler where the temperature may be 1,000° F.

Taking the previous example, but raising the superheat to 600°, then

Each pound of steam requires (600 — 387) 0·599, or 127·6 B.T.U., and each square foot passes 6 (1,000 — 600), or 2,400 B.T.U.

Then the heating surface per H.P. = 127·6 × 12 ÷ 2,400, or 0·638 square foot

It will be seen by the above that for variable service an independently fired superheater possesses many advantages, and when oil-firing is the vogue this could be done, as the waste heat from it could be utilised for feed-heating or other equally useful purposes.

The claims for superheating in practice on shipboard have been set out by several engineers at the Institution of Naval Architects, in the *Transactions* of which they may be read with advantage.

Mr. H. Gray produced in 1914 the following, as accomplished with a little more than 1 square foot of superheating surface per I.H.P. :—

MR. H. GRAY'S EXPERIMENTS WITH SUPERHEATED STEAM ON SHIPBOARD, 1914.

Steam Chests of	S.S. "P. A." Triple Expansion Engines.				S.S. "P. L." Quadruple Expansion Engines.			
	Gauge Steam Pressure.	Temp. if Saturated Steam.	Actual Temp. as Recorded	Amount of Superheat.	Gauge Steam Pressure.	Temp. if Saturated Steam.	Actual Temp. as Recorded	Amount of Superheat.
	Lbs.	° F.	° F.	° F.	Lbs.	° F.	° F.	° F.
High-pressure cylr.,	160	370°	550°	180°	206	389°	600°	211°
1st intrmdt. cylr.,	53	300°	400°	100°	97	335°	460°	125°
2nd intrmdt cylr.,	..	..	..	..	38	284°	290°	6°
Low-pressure cylr.,	10	240°	..	..	9	237°	220°	−17°

N.B.—Cuts-off—H.P. cylinder, 73 per cent.; 1 M.P., 70 per cent.; 2 M.P., 70, and L.P., 70 per cent.

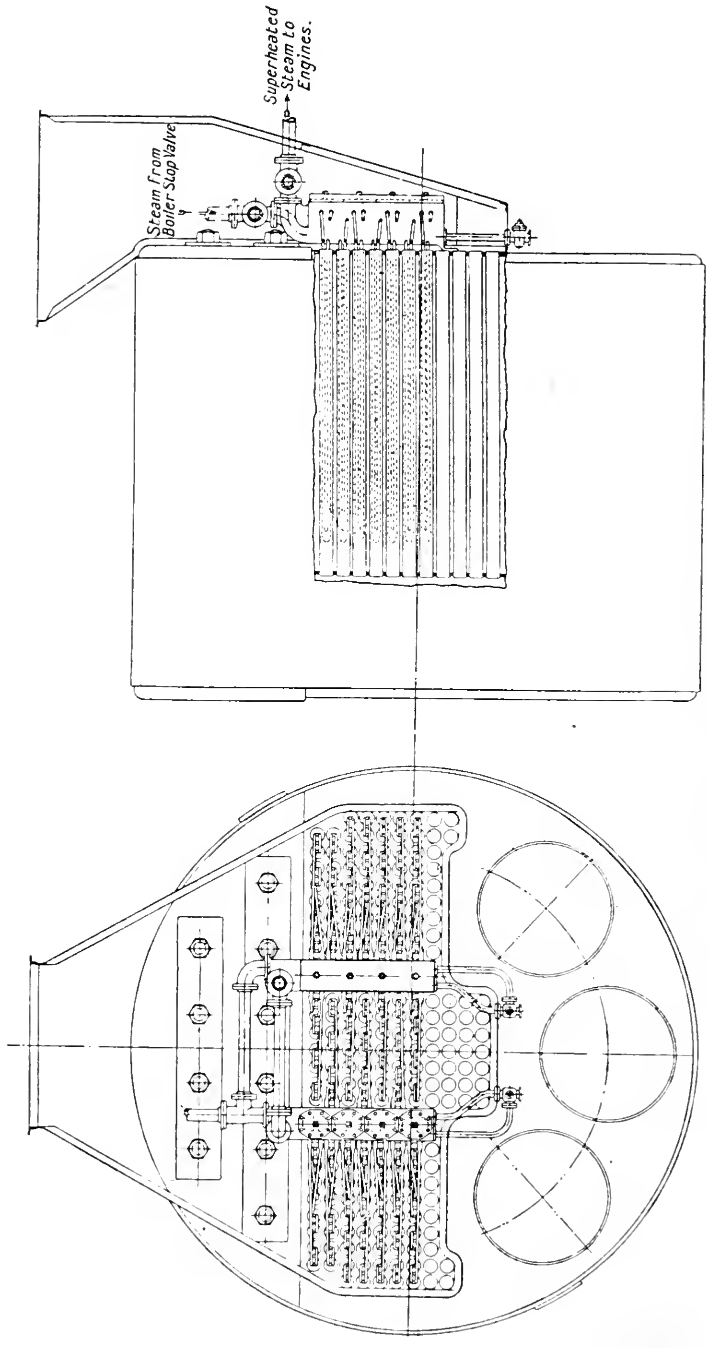


Fig. 334.—Arrangement of Superheater for Three-Furnace Boiler.

The s.s. "Port Augusta," with triple-expansion engines, consumed 1·6 lbs. of coal without superheat and 1·4 lbs. with superheat on an Australian voyage, while a similar ship with quadruple-expansion engines and steam superheated to 600° consumed only 1·29, the fuel being in each case the ordinary coal procurable at the ports of call.

On one experimental voyage with the latter ship using only good English coal, the consumption was as low as 1·043 lbs., and the average of five similar voyages with such coal showed it to be 1·15 lbs.

Mr. Gray claims as the result of his observations that with triple-expansion engines the gain by superheating to about 550° F. is 0·2 lb. per I.H.P., or $12\frac{1}{2}$ per cent., and with quadruple-expansion engines it is 0·19 lb., or 14·2 per cent.

The N. D. Lloyd steamer, "Columbus," of 15,000 I.H.P., with four-crank triple-expansion engines using superheated steam, had a consumption of 1·05 lbs. of coal per I.H.P.-hour for main engines and their own auxiliaries, and 1·20 lbs. consumption for all purposes throughout the ship.

In another large ship with quadruple engines of 21,650 I.H.P. the consumption of saturated steam was 12·71 lbs., and of coal 1·47 lbs., while with superheating by 200° F. it was reduced to 10·4 lbs. and 1·257 lbs., or 18·2 per cent. less. In a sister ship driven by turbines the consumption without superheat was 11·86 lbs. of steam and 1·385 lbs. of coal, as against 10·9 lbs. and 1·292 with 100° of superheat, or a gain of 8·1 per cent.

The s.s. "Jupiter" with triple-expansion engines indicating 2,600 H.P. supplied with steam at 185 lbs., having 280° F. of superheat, so that its temperature was 661° F., the consumption is only 1·0912 lbs. of Newcastle coal per I.H.P.-hour, which is exceedingly low for such engines.

The Robinson Superheater (Fig. 334), which is now largely used on shipboard, consists of a series of quite small diameter tubes in pairs joined by a special junction piece at their inner ends, so as to lie side by side for insertion into the boiler tubes; their outer ends are expanded into holes in special headers (Fig. 335), so that the steam flows from the boiler through them to the main steam pipe leading to the engine. The headers are grouped and secured near the front tube plate, so as not to obstruct gas flow from the tubes more than necessary. In this way the necessary high temperature for superheat is obtained without so high a temperature in the smoke-box as would otherwise be imperative. With such a large number of these small tubes the velocity of flow through them is comparatively small, so that the drop in pressure is not great.

It was common practice, and will be so now that for the superheater there is a separate circuit of pipes and valves, so that it may be shut out of action altogether or the portion of steam passing through it regulated so that the temperature may not, at any time as passed to the engines, be above that prescribed.

The Cylinder Lubricant, if any is used, must be a mineral oil having a high boiling point, and although there is not much risk of internal combustion, such a thing has happened with disastrous results, it should have, therefore, a high flash point too. In a general way, of course, there is no free oxygen present in a steam cylinder, but decomposition of water can take place and a supply of oxygen thereby provided. A very soft, greasy graphite has been used as a lubricant for internal purposes, and in moderation

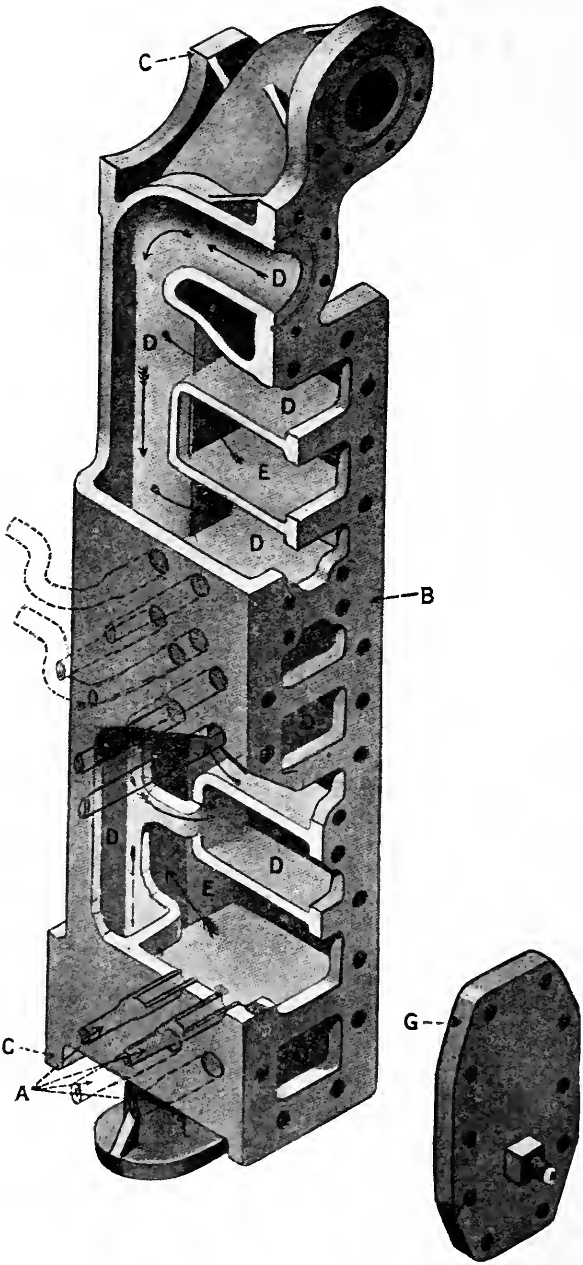


Fig. 335.—Sectional Perspective View of "Robinson" Superheater Header.

is said to be satisfactory with superheated steam; so much so that in some foreign ships superheat is much higher than is usual in British practice, the steam temperature being nearly 700° F., or over the melting point of lead, and not much below that of zinc; consequently no white metal can be used in the stuffing-boxes, and only special bronzes employed with safety there. Probably very soft grey cast iron would be best for neck rings, packing rings, and glands, as the graphite in it would provide the lubricant necessary.

TABLE CXI.—WEIGHT OF SUPERHEATED STEAM IN LBS. PER MINUTE THROUGH SMOOTH PIPES 240 DIAMETERS LONG, WITH A DROP IN PRESSURE OF 1 LB.

Gauge Pressure.	Diameter of Bore of Pipe in Inches.									
	1	2	3	4	5	6	7	8	9	10
100	5·25	26·0	64·2	118·5	196	293	400	535	690	863
120	5·63	27·9	69·9	127·2	210	315	433	574	740	925
140	6·00	29·7	73·2	135·0	223	334	458	609	786	980
160	6·31	31·3	77·4	142·0	235	352	483	640	825	1033
180	6·64	33·0	81·5	148·9	245	368	503	670	865	1080
200	6·95	34·5	85·7	155·8	256	383	525	698	903	1125
220	7·25	36·0	89·8	162·3	266	398	545	726	940	1172
240	7·55	37·5	93·8	168·4	275	412	563	752	983	1214

APPENDIX F

Board of Trade Rules for Boilers.

The Surveyor's Duty and Responsibility in Fixing Pressures.—The Surveyor is required by the Act to fix the limits of weight to be placed on the safety-valves of passenger steamships. The Surveyor having himself fixed the limits of the weight, is then required to declare, that in his judgment the boiler and machinery are sufficient for the service intended, and in good condition, and that they will be sufficient for twelve months, or such other period as he may, in his judgment, determine. For his guidance the following suggestions are given:—

Working Pressure to be Fixed by Calculation, &c.—The Surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, and according to the workmanship and material. The senior Engineer Surveyors should receive and report on any plans of boilers intended for passenger vessels that may be submitted in due course with the Form Surveys 6. They are not to report on any tracing or plan that is not accompanied by that form. When the Surveyor has received plans and tracings of new boilers, or of alterations of boilers, and has approved of them, he will of course be careful in making his examinations from time to time to see that they are followed in construction. When he has not had the plans submitted, but is called in to survey a boiler, he will then measure the parts, note the details of construction, and if necessary bore the plates to ascertain their thickness, &c., before he gives his declaration. And in the event of any novelty in construction, or of any departure from the practice of staying and strengthening noted in these regulations, he should report full particulars to the Board of Trade before fixing the working pressure.

The Surveyor cannot declare a boiler to be safe of whose construction, material, and workmanship he is not fully informed. He should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called in to survey until after it is completed and fixed in the ship.

Girders for Flat Surfaces.—When the tops of combustion boxes, or other parts of a boiler, are supported by solid rectangular girders, the following formula, which is used by the Board of Trade, will be useful for finding the working pressure to be allowed on the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers, that the ends are properly bedded to the edges of the tube plate and the back plate of the combustion box:—

$$\frac{C \times d^2 \times T}{(W - P) D \times L} = \text{Working pressure.}$$

W = Width of combustion box in inches.
P = Pitch of supporting bolts in inches.
D = Distance between the girders from centre to centre in inches.
L = Length of girder in feet.
d = Depth of girder in inches.
T = Thickness of girder in inches.

N = Number of supporting bolts.

$C = \frac{N \times 1,320}{N + 1}$ when the number of bolts is odd.

$C = \frac{(N + 1)\ 1,320}{N + 2}$ when the number of bolts is even.

The working pressure for the supporting bolts, and for the plate between them, shall be determined by the rule for ordinary stays and plates.

Plates for Flat Surfaces.—The pressure on plates forming flat surfaces will be easily found by the following formula:—

$$\frac{C \times (T + 1)^2}{S - 6} = \text{Working pressure.}$$

T = Thickness of the plate in sixteenths of an inch.
S = Surface supported in square inches.
C = Constant according to the following circumstances:—

C = 240 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and outside doubling-strips of the same thickness as the plates they cover, and not less in width than two-thirds of the pitch of the stays. The strips to be riveted on the plate.

NOTE.—When doubling-plates cover the whole of the flat surface, the case should be submitted for the consideration of the Board.

C = 210 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and washers, the latter on the outside of boiler, being at least two-thirds the pitch of the stays in diameter, and the same thickness as the plates they cover. These washers to be riveted on the plate.

C = 165 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of plates and outside washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover.

C = 150 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts only on both sides of plates.

C = 112·5 when tube plates are not exposed to the direct impact of heat or flame, and the stays are fitted with nuts.

C = 77 when the tube plates are not exposed to the direct impact of heat or flame, and the stay tubes are screwed and expanded.

C = 77 when the plates are not exposed to the impact of heat or flame, and the stays are screwed into the plates and riveted over.

C = 75 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts on both sides of plates, and outside washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plate they cover.

C = 67·5 when the plates are exposed to the impact of heat or flame, and steam in contact with the plate, and the stays fitted with nuts only.

C = 100 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plate and fitted with nuts.

C = 66 when the plates are exposed to the impact of heat or flame, with water in contact with the plate, and the stays screwed into the plate, having the ends riveted over to form substantial heads.

C = 39·6 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, with the stays screwed into the plate, and having the ends riveted over to form substantial heads.

When a circular flat end is bolted or riveted to a cylindrical shell, S in the formula may be taken as the area of the square inscribed in the circle passing through the centres of the bolts or rivets securing the end, provided the angle ring or flange is of sufficient thickness.

Compressive Stress on Steel Tube Plates.—

$$\frac{(D - d)\,T \times 28{,}000}{W \times D} = \text{Working pressure.}$$

D = Least horizontal distance between centres of tubes in inches.
d = Inside diameter of ordinary tube in inches.
T = Thickness of tube plate in inches.
W = Width of combustion box in inches between tube plate and back of fire box, or distance between combustion box tube plates when boiler is double-ended and the box common to the furnaces at both ends.

Cylindrical Boilers.—When cylindrical boilers are made of the best material, with all the rivet holes drilled in place, and all the seams fitted with double butt straps, each of at least five-eighths the thickness of the plates they cover, and all the seams at least double-riveted with rivets having an allowance of not more than 87·5 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of construction, then 4·5 may be used as the factor of safety. The tensile strength of the material is to be taken as equal to 27 tons per square inch when mild steel is used, tested in accordance with the Rules (p. 442). When plates of higher tensile strength than 27 tons are used, the actual strength, as found on test, may be substituted in the following Rule. The boilers must be tested by hydraulic pressure to twice the working pressure in the presence and to the satisfaction of the Surveyor:—

To ascertain the strength of shell, the relative sectional areas of plate and rivet must first be determined by the following formulæ:—

$$\frac{(\text{Pitch} - \text{dia. of rivet}) \times 100}{\text{Pitch}} = \begin{cases}\text{Percentage of strength of plate at joint}\\ \text{as compared with solid plate.}\end{cases}$$

For maximum permissible pitch of rivets, &c., see formulæ and sketches at end of this section. (Pp. 277 to 281, Seaton and Rounthwaite's *Pocket-Book of Marine Engineering.*)

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{Pitch} \times \text{thickness of plate}} = \begin{cases}\text{Percentage of strength}\\ \text{of rivets as compared}\\ \text{with solid plate.}\end{cases}$$

If the rivets are in double shear, multiply the percentage thus found by 1·875

In consequence of the low shearing strength of steel rivets, the Board require that in all types of joint the *nominal* rivet section shall be not less than $\frac{27}{23}$ of the net plate section; thus, in order that rivet section may be considered to have the same strength as plate section their relation must be:—

In lap joints—

$$(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 23 = (\text{Pitch} - \text{diameter of rivet}) \times \text{thickness of plate} \times 27.$$

And in butt joints—

$$(\text{Area of rivet} \times \text{No. of rows of rivets} \times 1{\cdot}875 \times 23 = (\text{Pitch} - \text{diameter of rivets}) \times \text{thickness of plate} \times 27.$$

But when plate of a higher minimum tensile strength than 27 tons is used, the actual strength in tons should be used instead of the multiplier 27.

The working pressure per square inch that may be allowed on the safety valves is then given by—

$$\text{Working pressure} = \frac{S \times \% \times 2\,T}{D \times F},$$

Where S = Tensile strength of material in lbs. per square inch.
% = One of the two percentages, found by Rules above divided by 100.
T = Thickness of plate in inches.
D = Inside diameter of boiler in inches (inside diameter of outer strake, if any).
F = Factor of safety from following Table if % refers to plate section.
F = 4·5 if % refers to rivet section.

The smaller of the two results to be taken.

Various penal additions are provided in the rules for cases in which the workmanship is of inferior character.

If it is proposed to use steel for superheaters, particulars should be submitted to the Board for consideration.

TABLE CIX.—Board of Trade Factors of Safety.

When all rivet holes are drilled in place after bending; *all* seams fitted with double butt-straps, each at least five-eighths the thickness of the plates they cover; all seams at least double riveted; and boilers open to inspection during construction, -	F = 4·5
To be added when circumferential seams are lap and double riveted, -	·1
To be added when longitudinal seams are lap and double riveted, -	·2
To be added when longitudinal seams are lap and treble riveted, -	·1
To be added when boiler is of such length as to fire from both ends, unless middle circumferential seams are lap and treble riveted, -	·3

Compensating rings should be fitted around all manholes and openings, of at least the same effective sectional area as the plate cut out, and in no case should rings be of less thickness than plate to which they are attached. Manholes in shells of cylindrical boilers should have their shorter axis placed longitudinally.

It is very desirable that compensating rings round openings in flat surfaces be of L or T iron.

The neutral parts of shells under steam domes must be efficiently stiffened and stayed.

Board of Trade Rules for Cylindrical Boiler Shells.

JOINTS WITH DRILLED HOLES.

Formulæ for ordinary chain riveted and ordinary zig-zag riveted joints, and for joints of these descriptions, when every alternate rivet in the outer or in the outer and inner rows have been omitted :—

Let E = distance from edge of plate to centre of rivet in inches.
V = distance between rows of rivets in inches.
V_1 = distance between inner and middle row of rivets in inches, for chain and zig-zag treble riveted butt joints with alternate rivets omitted in outer rows.
B = boiler pressure in lbs. per square inch.
C = 1 for lap or single butt joints.
C = 1·875 for double butt joints.
d = diameter of rivets in inches.
D = inside diameter of boiler in inches.
F = factor of safety for shell plates.
n = number of rivets in one pitch.
p_D = diagonal pitch in inches.
P_D = diagonal pitch in inches between inner and middle rows of rivets in inches, for zig-zag treble riveted butt joints with alternate rivets omitted in outer rows
p = greatest pitch of rivets in inches.
r = percentage of plate left between holes in greatest pitch.
R = percentage of rivet section.
R_1 = percentage of combined plate and rivet section.
S = tensile strength of material in lbs. per square inch of section.
S_1 = tensile strength of plates in tons.
T = thickness of plate in inches.
T_1 = thickness of each butt strap in inches.
% = least value of r, R, R_1, as the case may be, divided by 100.

ORDINARY CHAIN AND ZIG-ZAG RIVETED JOINTS.

Iron plates and iron rivets, or steel plates and steel rivets :—

$$\frac{100(p - d)}{p} = r.$$

Steel plates and steel rivets :—

$$\frac{100 \times 23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times p \times T} = R.$$

Given C, d, F, n, T, to find p, so that r and R are equal.

Steel plates and steel rivets :—

$$\frac{23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times T} + d = p.$$

Given C, F, n, T, r, to find p and d.

Steel plates and steel rivets :—

$$\frac{4 \cdot 5 \times S_1 \times r \times T}{23 \times (100 - r) \times \cdot 7854 \times n \times C \times F} = d.$$

$$\frac{100 \times 4 \cdot 5 \times S_1 \times r \times T}{23 \times (100 - r)^2 \times \cdot 7854 \times n \times C \times F} = p.$$

Steel plates and steel rivets when d is found first, then :—

$$\frac{100\,d}{100 - r} = p.$$

Steel plates and steel butt straps.

Double butt straps :—

$$\frac{5 \times T}{8} = T_1.$$

Single butt straps :—

$$\frac{9 \times T}{8} = T_1.$$

For Distance between Rows of Rivets, &c., for Double, Treble, and Quadruple Riveted Lap Joints; and Double and Treble Riveted Butt Joints with same Number of Rivets in Inner and in Outer Rows.

Steel plates and steel rivets :—

$$\frac{3 \times d}{2} = E.$$

Chain-riveted joints, not less than :—

$$2 \times d = V.$$

Zig-zag riveted joints :—

$$\frac{\sqrt{(11\,p + 4\,d)(p + 4\,d)}}{10} = V.$$

Diagonal pitch for zig-zag riveted joints :—

$$\frac{6\,p + 4\,d}{10} = p_D.$$

To determine the Working Pressure.

$$\frac{S \times \% \times 2\,T}{F \times D} = B.$$

Chain and Zig-zag Riveted Joints in which every alternate Rivet has been omitted in the Outer Row, or in the Outer and the Inner Rows.

Steel plates and steel rivets :—

$$\frac{100\,(p - d)}{p} = r.$$

Steel plates and steel rivets :—

$$\frac{100 \times 23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times p \times T.} = R.$$

Steel plates and steel rivets :—

$$\frac{100\,(p - 2\,d)}{p} + \frac{R}{n} = R_1.$$

When the Joints are fitted with Single or Double Butt Straps and the Number of Rivets in the Inner Row is double the Number in the Outer Row.

Steel plates and steel butt straps.

Double butt straps :—

$$\frac{5 \times T\,(p - d)}{8 \times (p - 2\,d)} = T_1.$$

Single butt straps:—

$$\frac{9 \times T(p - d)}{8 \times (p - 2d)} = T_1.$$

When the number of rivets in the inner row is the same as in the outer row.

Double butt straps:—

$$\frac{5 \times T}{8} = T_1.$$

Single butt straps:—

$$\frac{9 \times T}{8} = T_1.$$

For Distance between Rows of Rivets, &c., for Double and Treble Riveted Butt Joints with Alternate Rivets omitted in Outer Rows; and Treble Riveted Lap Joints and Butt Joints with alternate Rivets omitted in Outer and in Inner Rows.

Steel plates and steel rivets:—

$$\frac{3 \times d}{2} = E.$$

Chain-riveted joints:—

$$\left.\begin{array}{l} \dfrac{\sqrt{(11p + 4d)(p + 4d)}}{10} = V, \\ \text{or} \quad 2 \times d = V. \end{array}\right\} \text{The greatest of these two values of V to be used. } \textit{See Note below.}$$

For treble chain-riveted joint with alternate rivets omitted in outer row:—

$$2 \times d = V_1.$$

Zig-zag riveted joints:—

$$\sqrt{(\tfrac{11}{20}p + d)(\tfrac{1}{20}p + d)} = V.$$

Diagonal pitch:—

$$\tfrac{3}{10}p + d = p_D.$$

For treble zig-zag riveted joint with alternate rivets omitted in outer row:—

$$\frac{\sqrt{(11p + 8d)(p + 8d)}}{20} = V_1.$$

Diagonal pitch:—

$$\frac{3p + 4d}{10} = P_D.$$

To determine the Working Pressure.

$$\frac{S \times \% \times 2T}{F \times D} = B.$$

NOTE.—The minimum value of V or V_1 for chain-riveted joints is given as $2d$, but $\dfrac{4d + 1}{2}$ is more desirable.

Maximum Pitches for Riveted Joints.

T = thickness of plate in inches.
p = maximum pitch of rivets in inches.
C = constant applicable from the following table:—

TABLE CX.

Number of Rivets in One Pitch.	Constants for Lap Joints.	Constants for Double-Butt Strap Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	...	6·00

$$(C \times T) + 1\tfrac{5}{8} = p.$$

When the work is first-class, such pitches may be adopted so far as safety is concerned; yet, in some cases, it may be well not to adopt the greatest pitch found by the formula. The maximum should *not*, however, exceed $10\frac{1}{2}$ inches with the thickest plates for boiler shells.

If, in any case, the pitch is found to exceed that arrived at by the foregoing formula, for the particular description of joint and thickness of plate, such pitches should *not* be passed, but, in all cases, reported.

Note.—If the edge of the plate or butt strap is cut away or *vandyked* so as to follow the line of rivets, the pitch of rivets in the outer rows may exceed $10\frac{1}{2}$ inches.

APPENDIX G.

Lloyd's Rules for Determining the Working Pressure to be Allowed in New Boilers.

CYLINDRICAL SHELLS OF STEEL BOILERS.

The strength of cylindrical shells of steel boilers is to be calculated from the following formula :—

$$\frac{C \times (T - 2) \times B}{D} = \text{working pressure in lbs. per square inch.}$$

Where D = mean diameter of shell in inches.

T = thickness of plate in sixteenths of an inch.

C = 22, when the longitudinal seams are fitted with double butt straps of equal width.

C = 21·25, when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets.

C = 20·5, when the longitudinal seams are lap joints.

If the minimum tensile strength of shell plates is other than 28 tons per square inch, these values of C may be correspondingly increased.

B = the least percentage of strength of longitudinal joint,* found as follows :—

For plate at joint, $B = \frac{p - d}{p} \times 100.$

„ rivets „ $B = \frac{n \times a}{p \times t} \times 85$, where steel rivets are used.

$B = \frac{n \times a}{p \times t} \times 70$, „ iron „

Where p = pitch of rivets in inches.

t = thickness of plate in inches.

d = diameter of rivet holes in inches.

n = number of rivets used per pitch in the longitudinal joint.

a = sectional area of rivet in square inches. In case of rivets in double shear $1{\cdot}75\,a$ is to be used instead of a.

Note.—For the shell plates of superheaters or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the coefficients should be $\frac{2}{3}$ of those given in the preceding tables.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

* The inside butt strap to be at least $\frac{5}{8}$ of the strength of the longitudinal joint.

STAYS.

The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely :—

Iron Stays.—For stays not exceeding $1\frac{1}{2}$ inches smallest diameter, and for all stays which are welded, 6000 lbs. per square inch; for unwelded stays above $1\frac{1}{2}$ inches smallest diameter, 7500 lbs. per square inch.

Steel Stays.—For *screw* stays not exceeding $1\frac{1}{2}$ inches smallest diameter, 8000 lbs. per square inch; for *screw* stays above $1\frac{1}{2}$ inches smallest diameter, 9000 lbs. per square inch. For other stays not exceeding $1\frac{1}{2}$ inches smallest diameter, 9000 lbs. per square inch, and for stays exceeding $1\frac{1}{2}$ inches smallest diameter, 10,000 lbs. per square inch. No steel stays are to be welded.

Stay Tubes.—The stress is not to exceed 7500 lbs per square inch.

FLAT PLATES.

The strength of flat plates supported by stays is to be taken from the following formula :—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where T = thickness of plate in sixteenths of an inch.

P^2 = square of pitch in inches. If the pitch in the rows is not equal to that between the rows, then the mean of the squares of the two pitches is to be taken.

C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads.

C = 100 for iron or steel plates above $\frac{7}{16}$ thick, fitted with screw stays with riveted heads.

C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts.

C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts.

C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts.

C = 140 for iron plates, fitted with stays with double nuts.

C = 150 for iron plates, fitted with stays with double nuts and washers outside the plates, of at least $\frac{1}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates.

C = 160 for iron plates, fitted with stays with double nuts and washers riveted to the outside of the plates, of at least $\frac{2}{5}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates.

C = 175 for iron plates, fitted with stays with double nuts and washers riveted to the outside of the plates, when the washers are at least $\frac{2}{3}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{2}{3}$ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those in combustion chambers, the values of C may be increased as follows :—

C = 140	increased to	175.
C = 150	„	185.
C = 160	„	200.
C = 175	„	220.
C = 190	„	240.

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times \left(T + \frac{t}{2}\right)^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where t = thickness of doubling plates in sixteenths, and C, T, and P are as above.

NOTE.—In the case of front plates of boilers in the steam space, these numbers should be reduced 20 per cent., unless the plates are guarded from the direct action of the heat.

For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where T = the thickness of the plates in sixteenths of an inch.
P = the *mean* pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes, the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch.}$$

Where P = the horizontal distance from centre to centre of the bounding rows of tubes; and
C = 120, where the stay tubes are pitched with two plain tubes between them and are not fitted with nuts outside the plates.
C = 130, if they are fitted with nuts outside the plates.
C = 140, if each alternate tube is a stay tube not fitted with nuts.
C = 150, if they are fitted with nuts outside the plates.
C = 160, if every tube in these rows is a stay tube and not fitted with nuts.
C = 170, if every tube in these rows is a stay tube and each alternate stay tube is fitted with nuts outside the plates.

The thickness of tube plates of Combustion Chambers, in cases where the pressure on the top of the chambers is borne by these plates, is not to be less than that given by the following rule :—

$$T = \frac{P \times W \times D}{1750 \times (D - d)}.$$

Where P = working pressure in lbs. per square inch.
W = width of Combustion Chamber between plates in inches.
D = horizontal pitch of tubes in inches.
d = inside diameter of plain tubes in inches.
T = thickness of tube plates in sixteenths of an inch.

GIRDERS.

The strength of girders supporting the tops of combustion chambers and other flat surfaces is to be taken from the following formula :—

$$\frac{C \times d^2 \times T}{(L - P) \times D \times L} = \text{working pressure in lbs. per square inch.}$$

Where L = width between tube plates, or tube plate and back plate of chamber.
P = pitch of stays in girders.
D = distance from centre to centre of girders.
d = depth of girder at centre.
T = thickness of girder at centre.
All these dimensions to be taken in inches.

Wrought Iron.

C = 6,000, if there is one stay to each girder
C = 9,000, if there are two or three stays to each girder.
C = 10,000, if there are four or five stays to each girder.
C = 10,500, if there are six or seven stays to each girder.
C = 10,800, if there are eight stays or above to each girder.

Wrought Steel.

C = 7,110, if there is one stay to each girder.
C = 10,660, if there are two or three stays to each girder.
C = 11,850, if there are four or five stays to each girder.
C = 12,440, if there are six or seven stays to each girder.
C = 12,800, if there are eight stays or above to each girder.

CIRCULAR FURNACES.

The strength of plain furnaces to resist collapsing to be calculated as follows :—

Where the length of the plain cylindrical part of the furnace exceeds 120 times the thickness of the plate, the working pressure is to be calculated by the following formula :—

$$\frac{1{,}075{,}200 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch.}$$

Where the length of the plain cylindrical part of the furnace is less than 120 times the thickness of the plate, the working pressure is to be calculated by the following formula :—

$$\frac{50 \times (300\,T - L)}{D} = \text{working pressure in lbs. per square inch.}$$

Where D = outside diameter of furnace in inches.
T = thickness of plates in inches.
L = length of plain cylindrical part in inches, measured from the centres of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the commencement of the curvature of the flanges of the furnace where it is flanged or fitted with Adamson rings.

In the furnaces referred to below, the formulæ given are applicable if the steel used has a tensile strength of not less than 26 nor more than 30 tons per square inch. If the material of furnaces has a less tensile strength than 26 tons per square inch, then, for each ton per square inch which the minimum tensile strength falls below 26, the coefficient is to be correspondingly decreased by $\frac{1}{26}$ part.

The strength of corrugated furnaces made of steel, on Fox's, Morison's, Deighton's, or Beardmore's plan, and of the Leeds Forge bulb furnace, to be calculated from

$$\frac{1259 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of Brown's cambered, and improved Purves' furnaces (with ribs 9 inches apart) to be calculated from the following formula :—

$$\frac{1160 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of spirally corrugated furnaces is to be calculated from the following formula :—

$$\frac{912 \times (T - 2)}{D} = \text{working pressure in lbs. per square inch.}$$

Where T = thickness of plate in sixteenths of an inch; and
D = outside diameter of corrugated furnaces, in inches; or *smallest* outside diameter, in inches, of Brown's cambered, improved Purves', and Leeds Forge bulb furnaces.

The strength of Holmes' patent furnaces, in which the corrugations are not more than 16 inches apart from centre to centre, and not less than 2 inches high, to be calculated from the following formula :—

$$\text{Working pressure in lbs. per square inch} = \frac{945 \times (T - 2)}{D}.$$

Where T = thickness of plain portions of furnace in sixteenths of an inch.
D = outside diameter of plain parts of the furnace in inches.

APPENDIX H.

RULES OF THE BRITISH CORPORATION FOR THE WORKING PRESSURE, THICKNESSES OF PLATES, AND SIZES OF STAYS FOR STEEL BOILERS.

THE rules following are intended to apply to the construction of steel boilers; where boilers are to be made of iron, they will be specially considered by the committee.

Strength Calculations.—The sizes and arrangement of the different parts for a given working pressure, or the working pressure suitable to a given size and arrangement of material, may be found from the following formulæ and rules:—

Cylindrical Shells—

$$\frac{C \times (T - 1) \times E}{D} = W.$$

Where C = 20·4, when the longitudinal seams are fitted with double butt straps of equal width, and of thickness at least equal to that obtained from the formula for butt straps.

C = 19·7, when the double butt straps are of unequal width—*i.e.*, one strap not covering the outer row of rivets, with the thickness as before.

C = 19·0, when the longitudinal seams are lap joints.

T = thickness of shell plate in sixteenths of an inch.

E = the least percentage of strength of longitudinal joints, found as follows:—

For the plate at the joint, $E = \frac{p - d}{p} \times 100.$

For the rivets at the joint, $E = \frac{n \times a}{p \times t} \times \frac{r}{s} \times 100.$

Where p = pitch of rivets in inches.

d = diameter of rivet holes in inches.

n = number of rivets used per pitch.

a = sectional area of rivets in square inches.

t = thickness of plate in inches.

r = 23 for steel.

s = the minimum tensile strength of plate.

Where rivets are in double shear, 1·875 a is used instead of a.

D = mean diameter of shell in inches.

W = working pressure in lbs. per square inch.

NOTE.—The constant, C, is to be used with steel of the minimum strength of 28 tons per square inch, as required by the rules (see p. 810); when a higher minimum strength is guaranteed, the constant may be proportionately increased.

Double Butt Straps.—When all the rows of rivets in the butt are of the same pitch, each strap must be at least five-eighths of the thickness of the shell plate, and, when the pitch of the outer row of rivets is twice that of the centre row, the thickness of each plate is to be in accordance with the following formula :—

$$T_1 = T\,\frac{5\,(p - d)}{8\,(p - k\,d)}.$$

Where T_1 = thickness of strap in sixteenths of an inch.
T = thickness of shell in sixteenths of an inch.
p = pitch of rivets in inches.
d = diameter of rivet holes in inches.
k = ratio of pitch of rivets in outer to that in inner row.

Flat Surfaces, supported by Stays—

$$\frac{C \times T^2}{P^2 + p^2} = W.$$

Where C = 195 for plates fitted with screwed stays having riveted heads.
C = 265 for plates fitted with screwed stays and nuts.
C = 345 for plates fitted with stays and double nuts.
C = 370 for plates with stays, double nuts, and washers outside. The washers to be at least half the thickness of the plate, and in diameter equal to one-third the pitch of stays.
C = 400 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be at least half the thickness of the plate, and in diameter equal to two-fifths the pitch of the stays.
C = 450 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be the same thickness as the plate, and in diameter equal to two-thirds the pitch of the stays.
C = 480 for plates fitted with stays, double nuts, and doubling strips outside riveted to the plates. The strips to be the same thickness as the plate, and in width equal to two-thirds the pitch of the stays.
T = thickness of plate in sixteenths of an inch.
P = greatest pitch of stays.
p = least pitch of stays.
W = working pressure in lbs. per square inch.

Flat plates having doublings, at least two-thirds of their thickness, riveted to them.

$$\frac{C \times \left(T + \frac{t}{2}\right)^2}{P^2 + p^2} = W.$$

Where t = thickness of doubling plates in sixteenths of an inch.
C, T, P, p, and W, as before.

NOTE.—For front plates in the steam space, which are not protected against the direct action of the flame, the constants given above are to be reduced 20 per cent.

Tube Plates, with Tubes in Nests.—

$$\frac{275 \times T^2}{P^2 + p^2} = W.$$

Where T and W are as before.

P = greatest pitch of stay tubes from centre to centre in inches.
p = least pitch of stay tubes from centre to centre in inches.

For wider spaces, *v.* p. 904, *ante*).

When girders are fitted to the tops of combustion chambers, the thickness of the tube plates must be found from the formula—

$$\frac{W \times L \times P}{(P - d) \times 1{,}800} = T.$$

Where W = working pressure in lbs. per square inch.
L = width of combustion chamber over the plates in inches.
P = horizontal pitch of tubes in inches.
d = inside diameter of plain tube in inches.
T = thickness of tube plate in sixteenths of an inch.

Stays supporting Flat Surfaces—

$$\sqrt{\frac{S \times W}{C}} + \frac{1}{8} = D.$$

Where S = surface in square inches, supported by the stay.
W = working pressure in lbs. per square inch.
D = effective diameter of stay in inches.
C = 8,000 for steel screwed stays and iron if tested to $21\frac{1}{2}$ tons with 27 per cent. elongation.
C = 6,500 for iron screwed stays.
C = 8,900 for longitudinal steel stays.
C = 7,000 for longitudinal iron stays.
C = 5,000 for welded iron stays.

Stay Tubes are not to be subjected to a greater stress than 7,500 lbs. per square inch.

Circular Furnaces.—Thickness of plain furnaces and of furnaces with Adamson rings pitched more than 20 inches apart—

$$\frac{51{\cdot}5}{D}\{18{\cdot}75\,T - (L \times 1{\cdot}03)\} = W.$$

Thickness of bulb furnaces made by the Leeds Forge Company (Suspension)—

$$\frac{1{,}250 \times (T - 2)}{D} = W.$$

Thickness of furnaces with Adamson rings pitched not more than 20 inches apart, or corrugated, ribbed, and suspension furnaces (Fox, Purves, Morison, Deighton, and Brown's cambered)—

$$\frac{1{,}160 \times (T - 2)}{D} = W.$$

Thickness of Farnley spiral furnace and Holmes' furnace—

$$\frac{950 \times (T - 2)}{D} = W.$$

Where T = thickness of plate in sixteenths of an inch.
L = length of furnace or pitch of stiffening rings in inches.
D = smallest outside diameter of furnace in inches.
W = working pressure in lbs. per square inch.

Girders for Combustion Chamber Tops—

$$\frac{C \times d^2 \times T}{D \times L^2} \times \frac{(n + 1)^2}{n(n + 2)} = W, \text{ when } n = 2, 4, \text{ or } 6.$$

$$\frac{C \times d^2 \times T}{D \times L^2} \quad . \quad . \quad . \quad . \quad = W, \text{ when } n = 1, 3, \text{ or } 5.$$

Where C = 14,500 for steel and 13,180 for iron.
d = depth of girder at the centre in inches.
T = thickness of girder in inches.
D = distance from centre to centre of girders in inches.
L = length from tube plate to tube plate, or from tube plate to back of combustion chamber, in inches.
n = number of stays fitted with each girder.

General Construction.

1. Each boiler must have at least one glass water gauge, two test cocks, and one steam pressure gauge. Double-ended boilers are to have these fittings at each end. One salinometer cock must also be fitted to each boiler. Where the water gauge pillar is attached by pipes to the steam and water spaces, cocks or valves should be fitted to the shell of the boiler at the ends of these pipes.

2. A stop-valve is to be fitted to each boiler, so that any one of a series of boilers may be worked independently if required. The neck of the stop-valve to be as short as possible. All boiler and engine stop-valves to be tested to at least twice the working pressure.

3. Two safety-valves will be required for each main boiler, and they must be tested under steam to the satisfaction of the surveyors, and set to a pressure not more than 3 per cent. in excess of the intended working pressure. The combined area of the valves is to be sufficient to prevent the steam accumulating to more than 10 per cent. of the working pressure during fifteen minutes full firing, with main engines stopped. If the boilers be supplied with forced draught, the valve area must be increased so that the same conditions may be met.

4. Easing gear is to be provided, and so arranged that the safety-valves on any one boiler may be lifted simultaneously, without interfering with those on any other boiler.

5. Surface and bottom blow-off cocks should be fitted to the boiler, in addition to the cock on the hull of the vessel, and all cocks must have

spigots extending through the hull plating, with a plate flange round same on the outside.

6. It is recommended that manhole doors in boilers be not less than 16 inches by 12 inches.

7. Upon completion, the boilers and superheaters are to be tested by hydraulic pressure to twice the intended working pressure, and, after being placed in position in the vessel, they must be efficiently secured by brackets and stays to prevent any fore-and-aft or athwartship movement. It is strongly recommended, because of the rapid corrosion which takes place in material near them, that the boilers be kept as high as possible above the floors or tank top, and that the under side of the boilers be efficiently insulated.

8. Donkey boilers need not have more than one safety-valve, provided the valve area be not less than ½ square inch for each foot of grate surface. In other respects, the requirements for donkey boilers are the same as for main boilers.

Tubes for water-tube boilers and superheaters to be not less in thickness than required for steam pipes, and tested to 1,000 lbs. per square inch (hydraulic), or to three times the working pressure, whichever is greater. Headers also to be tested to three times the W.P. All other parts to double W.P.

APPENDIX I.

Bureau Veritas Rules for Boiler Shells, Working Pressure, or Thickness of Plates, and for Sizes of Stays.*

CIRCULAR SHELLS AND STEAM-HOLDERS WITH INTERNAL PRESSURE.

A riveted joint may fail through the tearing of the plate or butt strap between the rivets, the shearing of all the rivets, or by a combination of the two. The following formulæ apply to these several cases. The plate thickness and the diameter of rivets to be applied to have the highest values which each formula would give separately.

I. *Rupture through Plate.*

The formulæ for working pressure and plate thickness are in this case:—

$$\left.\begin{aligned} P &= \frac{2\,a\,R\,(t - 0{\cdot}04)}{D} \\ \text{and}\qquad t &= \frac{P\,D}{2\,a\,R} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad - \quad - \quad - \quad (\text{I.})$$

Where P = allowed working pressure, above atmosphere, in lbs. per square inch.

D = greatest inside diameter of boiler shell or steam-holder in inches.

t = thickness of shell plates in inches. $t - 0{\cdot}04$ inch represents the thickness left after a reduction of 0·04 inch through corrosion.

R = the tensile stress in lbs. per square inch, which will be allowed in the plate. The value of R will be the breaking strength divided by 4, the latter figure representing the factor of safety for the plate after it has been corroded away by 0·04 inch.

If the actual breaking strength happens to be known by tests carried out to the Administration's satisfaction, it may be applied for finding R; but when, as usual, it is not known, the value of R will be :—

For Steel.—The fourth part of the lower limit of tensile strength chosen by the designer, which, in such case, is to be stated when a boiler design is submitted for approval.

For Iron.—11,200 lbs. per square inch, corresponding with a tensile strength of 20 tons. A table annexed shows the values of 2 R for various tensile strengths.

a = ratio of the resistance of the plate left between the holes to that of the full plates. It will be determined from the following expression :—

$$a = \frac{p - d}{p}.$$

* *N.B.*—If a boiler is intended for a vessel belonging to a country where the law prescribes heavier scantlings than those required by the following paragraphs, the builders have, of course, to comply with the legal requirements.

Where p = pitch of rivets in outer row in inches.
d = diameter of rivet holes in inches, either the real diameter or a corrected one, according to the following clauses :—

1. When the rivet holes are drilled, or when, having been punched, they are afterwards drilled or rimered out so that the injured metal around is completely removed, the real diameter may be taken.
2. When the holes are simply punched, they will be considered as being $\frac{1}{4}$ inch larger in diameter than as punched.

TABLE CXI. SHOWING THE VALUES OF 2 R.

In Formulæ (I.) *and* (IV.) *for Various Tensile Strengths of the Material.*

Tensile Strength of Plates in Tons per Sq. Inch.	Value of 2 R.	Tensile Strength of Plates in Tons per Sq. Inch.	Value of 2 R.
32	35,800	24½	27,400
31	34,700	24	26,900
30	33,600	23½	26,300
29	32,500	23	25,800
28	31,400	22½	25,200
27	30,200	22	24,600
26½	29,700	21½	24,100
26	29,100	21	23,500
25½	28,600	20½	22,900
25	28,000	20	22,400

II. *Rupture through Rivets.*

In this case the following are the formulæ for finding the allowed working pressure or required rivet section :—

$$\left.\begin{aligned} P &= \frac{2 A s}{D l} \\ \text{and} \quad A &= \frac{P D l}{2 s} \end{aligned}\right\} \quad - \quad - \quad - \quad - \quad \text{(II.)}$$

Where P and D have the same meaning as before, and
l = the length in inches of the identical parts into which a riveted joint can be subdivided. In most cases, l is the pitch of the rivets in the outer rows (figs. 333 and 334). In general, it depends upon the system of joint adopted.
s = the maximum shearing stress in lbs. per square inch which will be allowed on the rivets. It will be the fourth part of the actual shearing resistance of the material, if known, from tests. If the actual shearing resistance of the rivet bars is not known, it will be assumed to amount to 0·8 of their tensile strength, and the value of s will be one-fifth of the lower tensile limit adopted by the designer, who is free to adopt from 24 to 29 tons steel.

For Iron.—9,000 lbs. per square inch, corresponding with a tensile strength of about 20 tons per square inch.

A = the total shearing surface in square inches of the rivets (that is, twice the area of the rivet hole when a rivet is in double shear). Only fifteen-sixteenths of the full area are to be taken when the riveting is done by hand.

III. *Combined Rupture through Plate and Rivets.*

This case is only to be examined when the outer row has a wider pitch than the inner ones.

The formula to be applied in this case is :—

$$P = \frac{2\,(B \times R + C \times S)}{D \times l} \quad . \quad . \quad . \quad . \quad \text{(III.)}$$

Where P, R, S, D, and l have the same meaning as before (*v.* figs. 333 and 334).

B = the sectional area in square inches of the plate on the portion, l, of the joint along the line of its supposed rupture, assuming that, in case the plate is liable to corrosion, its thickness has been reduced by 0·04 inch.

C = the total area of the rivets which are supposed to shear on the length, l, corrected, if required, in the same way as prescribed above.

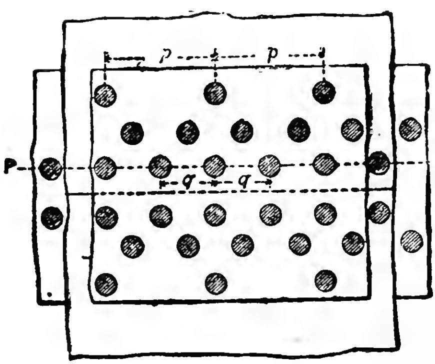


Fig. 336.

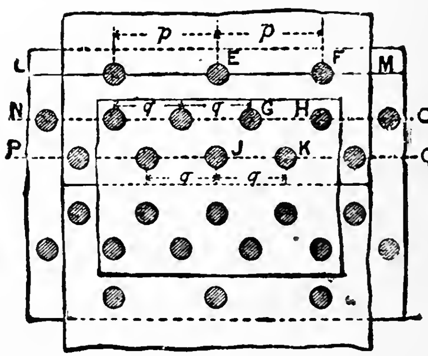


Fig. 337.

For a rivet in double shear, the resistance will be considered as being twice that of one in single shear.

IV. *Rupture through Butt Straps.*

Rupture may take place along one of the inner rows of rivets (see P, Q, figs. 333 and 334). The formula for this case, based on the same principle as (I.), are :—

$$\left.\begin{aligned} P &= \frac{2\,a\,R\,(t - 0{\cdot}04)}{D} \\ \text{and} \qquad t &= \frac{P \times D}{2\,a\,R} + 0{\cdot}04 \text{ inch.} \end{aligned}\right\} \quad . \quad . \quad . \quad \text{(IV.)}$$

Where P, D, and R have the same meaning as before.

t = thickness in inches of butt strap, or sum of thicknesses, if there are two straps. (The thickness, of course, not to be less than required for caulking.)

$$a = \frac{q - d}{q}.$$

Where q = pitch of rivets in the inner row in inches.

d = diameter in inches of the rivet holes in the inner row.

V. *Combined Rupture through Butt Straps and Rivets.*

Formula (III.) applies to this case, B being the section of the butt strap, or straps, along which rupture would take place.

Remarks.—No rivet holes to be nearer the edge of any plate than the diameter of the rivet. In zigzag riveting, the distance between the rows is to be such that no rupture through plate or butt strap is to be feared along the zigzag line.

When stays are bolted through the shell, they should be so arranged that they do not weaken the shell plates more than the riveted joints. If the resistance at the stay bolts is the smaller of the two, the plate's thickness shall be determined by it. It will be found from a formula the same as (I.), p and d applying to the stay bolts.

Shells of Superheaters.—The same mode of determining the working pressure or the thickness of shall plates will be followed for circular cylindrical superheaters, and the same formulæ may be used, but with the following alterations :—

1. When the plates are exposed to the direct action of the products of combustion, the values of R and S, as given for Cases I. and II., will have to be multiplied by 0·8 and the addition to the thickness of plates, on account of corrosion, will be increased from 0·04 to $\frac{3}{16}$ of an inch, to compensate for the corrosive action of the gases.

Formulæ (I.) become, therefore—

$$P = \frac{(1{\cdot}6\, a\, R\, t - \frac{3}{16})}{D},$$

and
$$t = \frac{P\, D}{1{\cdot}6\, a\, R} + \frac{3}{16} \text{ inch}.$$

2. When the plates are protected from the direct action of the products of combustion, R and S will be multiplied by 0·9, and the additional thickness for burning away will be $\frac{1}{8}$ inch.

Formulæ (I.) become in this case—

$$P = \frac{(1{\cdot}8\, a\, R\, t - \frac{1}{8})}{D}, \text{ and } t = \frac{P\, D}{1{\cdot}8\, a\, R} + \frac{1}{8} \text{ inch}.$$

FLAT PLATES.

The allowed working pressure or the thickness of flat plates is to be determined by the following formulæ :—

$$P = \frac{(t-1)^2}{a^2 + b^2} \times \frac{T}{C}, \text{ and } t = 1 + \sqrt{(a^2 + b^2)\frac{P\,C}{T}}.$$

Where P = allowed working pressure above atmosphere in lbs. per sq. inch.
t = thickness of plate in sixteenths of an inch.
a = pitch of stays in inches, in one row.
b = distance in inches between two rows of stay.

NOTE.—When plates are effectively stiffened by doubling plates well riveted thereto, and having a thickness, t, in sixteenths, the value, $t + \frac{t}{2}$ may be substituted for t in the formula.

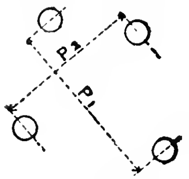


Fig. 338.

In case of irregular staying, such as in the annexed sketch, (fig. 335)—

$\frac{1}{4}(P_1 + P_2)^2$ shall be taken instead of $(a^2 + b^2)$.

T = tensile strength of the material in tons per square inch of the original section.

It is to be determined in the same way as for shell plates, that is—

For steel it will be equal to the lower limit of the tensile stress, which is to be stated on the drawing.

For iron, 21 tons per square inch.

C = a constant, the value of which depends upon the mode of staying, as follows ;—

C = 0·104 when the stays are screwed into the plates and riveted over.

C = 0·079 when the stays are screwed into the plates and fitted with outside nuts at both ends.

C = 0·062 when the stays are fitted with inside and outside nuts and washers, provided the diameter of the outside washers be at least 0·4 of the pitch between the rows of stays. The thickness of the washer to be at least $\frac{2}{3}$ that of the plate.

C = 0·055 when the stays are fitted with inside and outside nuts and washers, the outside washer being riveted to the plate and having $\frac{3}{4}$ of the plate's thickness and a diameter equal to 0·6 of the pitch between the rows of stays.

C = 0·050 when the outside washers are replaced by strips of plate, having a width of at least 0·6 of the distance between the rows of stays, with a thickness not less than $\frac{3}{4}$ that of the plate : the strip being well riveted to the plate.

For the values of $\frac{T}{C}$, see Table cxii.

When the plates are in contact with steam on one side and flame or hot gases on the other, the thickness is to be increased.

For instance, when, in return tube boilers, the top front plates are in no way protected from the hot gases, the working pressure or thickness will, in such a case, be determined by the formulæ :—

$$P = \frac{(t-2)^2}{a^2+b^2} \times \frac{0 \cdot 9\ T}{C},$$

and

$$t = 2 + \sqrt{(a^2+b^2)\frac{P\ C}{0 \cdot 9\ T}}.$$

When the said front plates are protected by a flame plate, no increase of thickness will be required.

When front plates are in two pieces, the lap should be double-riveted if the thicker plate is $\frac{1}{2}$ inch or above.

TABLE CXII.—VALUES OF $\frac{T}{C}$ IN THE FORMULÆ FOR FLAT PLATES.

Tensile Strength of Plate in Tons per Sq. Inch.	C = 0·104.	C = 0·079.*	C = 0·062.	C = 0·055.	C = 0·050.
20	$\frac{T}{C}$ = 192·3	$\frac{T}{C}$ = 253·1	$\frac{T}{C}$ = 322·6	$\frac{T}{C}$ = 363·6	$\frac{T}{C}$ = 400
21	202·0	265·8	338·6	381·8	420
22	211·5	278·4	354·8	400·0	440
23	221·1	291·1	371·0	418·0	460
24	230·8	303·8	387·0	436·2	480
25	240·4	316·4	403·2	454·4	500
26	250	329·1	419·4	472·6	520
27	259·6	341·8	435·4	490·8	540
28	269·2	354·4	451·6	509·0	560
29	278·8	367·1	467·8	527·2	580
30	288·4	379·7	483·8	545·4	600

* *N B.*—When plates are not subjected to flame or hot gases, C may be reduced from 0·079 to 0·066; or the values of $\frac{T}{C}$ in this column by 1·2.

STAYS.

The diameter of stays supporting flat surfaces is to be determined by the following formula :—

$$d = \tfrac{1}{8}\text{ inch} + \sqrt{\frac{Q}{300\ T}}.$$

Where d = effective diameter in inches (for instance, the diameter at bottom of thread in screw stays).

Q = total load on stay in lbs.

T = tensile strength of the material in tons per square inch.

For steel, this tensile strength will be the lower limit chosen by the boiler designer; for iron, it will be taken at 22 tons. In both cases the actual strength may be applied, if it is known from tests.

If the stays are not round, their cross-section must be such that the stress per square inch, caused by the load, Q, nowhere exceeds 5·75th part of the tensile strength, after deducting $\frac{1}{16}$ of an inch all round as an allowance for corrosion or wear.

In welded stays, the stress, as just described, will be reduced by 20 per cent. Welding of steel stays is only allowed for very mild qualities.

For high working pressures, such as used in triple-expansion engines, it is recommended to screw all stays into the plates they support, in addition to fitting them with nuts.

This also applies to stay tubes, with the exception that it is recommended that nuts should not be fitted in combustion chambers.

CIRCULAR FURNACES.

Plain Cylindrical Furnaces.

When made truly circular as possible, and of steel, not less than 26 tons tensile, the working pressure and the thickness of the plates may be calculated from the following formulæ:—

$$P = \frac{16{,}000\,t - 60\,L}{D},$$

and

$$t = \frac{P \times D + 60\,L}{16{,}000}.$$

Where t = the required thickness of plates in inches.
D = outside diameter of furnace in inches.
P = working pressure in lbs. per square inch (above atmosphere).
L = length of furnace in inches; or, if made or fitted with efficient rings, the length between the rings.

For iron or low tensile steel, constant must be reduced to 14,400.

Furnace plates should not exceed $\frac{13}{16}$ inch in thickness.

DOMED FURNACES IN VERTICAL BOILERS.

When the tops of furnaces in vertical boilers are portions of spheres the thickness must not be less than:—

$$t = \frac{P\,r}{8{,}000} + \cdot 15 \text{ for iron plates, or } t = \frac{P\,r}{9{,}000} + \cdot 15 \text{ for steel plates.}$$

CORRUGATED AND RIBBED FURNACES.

The plate thickness is to be found by the following formulæ:—

1. For corrugated furnaces—

$$T = \frac{P\,D}{1{,}260} + 2.$$

Where P = working pressure in lbs. per square inch.
T = thickness of plate in sixteenths of an inch.
D = outside diameter in inches measured on the top of the corrugations.

The formula applies to corrugations 6 inches long and $1\frac{1}{2}$ inches deep.

2. For ribbed furnaces, when manufactured to the satisfaction of the Administration—

$$T = \frac{PD}{1,160} + 2.$$

Where P and T are as above.

D = outside diameter of plain parts between the ribs in inches. The formula applies to ribs spaced 9 inches and projecting $1\frac{3}{8}$ inches, the difference between the greatest and the smallest diameters in any part of the furnace not exceeding $\frac{3}{1000}$.

3. For bulb furnaces—

$$T = \frac{PD}{1,260} + 2.$$

Where D is the outside diameter in inches measured between the bulbs.

The coefficients in the above formula apply to the case where the tensile strength of the material is 26 tons or above per square inch. When it is below 26 tons the coefficient is to be reduced $\frac{1}{26}$ for each ton below 26.

Combustion Chamber Girders.

The strength of girders on the tops of combustion chambers shall be determined as follows :—

$$P = \frac{C\,d^2\,t}{(W - p)\,D\,L}.$$

Where P = working pressure.
C = a constant found as under.
N = number of bolts in each girder.
d = depth of girder in inches.
t = thickness of girder in inches.
p = pitch of bolts in girder in inches.
L = length of girder between supports in inches.
D = distance between girders, centre to centre, in inches.
W = width of firebox, from tube plate to back plate, in inches.

$$C = \frac{13,200\,N}{N + 1} \text{ for odd numbers of bolts.}$$

$$C = \frac{13,200\,(N + 1)}{N + 2} \text{ for even numbers of bolts.}$$

GENERAL AS TO BOILER ARRANGEMENTS AND FITTINGS.

When two or more boilers are fitted, they should be arranged to work separately or independently of each other, either by stop-valves between the boilers and the common superheater, or by stop-valves between the separate superheaters and the main steam pipes.

Each boiler must be fitted with a separate pressure gauge. Double-ended boilers to have a pressure gauge at each end.

Steam pipes of donkey engines must be independent of the main pipes, so as to keep the steam off the main engines when only winches or other auxiliary engines are working.

The bottom blow-off to be arranged with one cock directly attached to the shell of the boiler, and another directly attached to the skin of the vessel. The surface blow-off must be similarly arranged. The main stop-valve to be so situated and fitted that it can be worked from the starting platform or the stokehole floor.

To protect the plating, the cocks on the ship's bottom should be fitted with spigots passing through the plating and through a flange on the outside. If this flange is of iron it must be galvanised.

Steam domes or superheaters, when placed in the uptake and exposed to the direct impact of the flame, will only be allowed as an exception, and must be efficiently protected by flame plates. In all cases it must be possible to examine efficiently the interior and the exterior of the domes or superheaters.

To prevent the boilers shifting in a transverse direction through the rolling of the vessel, or longitudinally in case of collision, they must be properly secured in their seats.

All manholes to be fitted with compensating rings.

At least two safety-valves of an approved design must be fitted to each main boiler.

Their total area is given by the following formula :—

$$A = \frac{8 \cdot 7}{\sqrt[4]{(P - 18)^3}}.$$

Where A = sectional area of safety-valve in square inches per square foot of grate surface.

P = working pressure of boiler in lbs. per square inch.

When forced draught is provided for, the area is to be increased in proportion to the increased evaporative power of the boilers.

Suitable arrangements and gear to be fitted in connection with the safety-valves, whereby they may be lifted from the deck as well as from the stokehole floor.

If it be practicable to isolate a superheater communicating simultaneously with two or more boilers, it will be necessary to furnish it with a safety-valve of suitable dimensions.

APPENDIX J.

Admiralty Rules for Steel Boilers and Materials and their Tests.

Admiralty rule for working pressure provides that the water-pressure test shall not exceed $\frac{4}{9}$ of the ultimate strength of the shell, or a factor of safety of $2\frac{1}{4}$, when subjected to water test pressure, and the working steam pressure is fixed at 90 lbs. below the test pressure, which is called their "constant margin" of safety for all pressures.

SUPERVISION OF BOILER WORK.

The following instructions to boiler-maker overseers are those usually given in Admiralty specification :—

The boilers will be subject to the supervision of an overseer, who will be directed to attend on the premises of the contractors during the progress of the work on the boilers, to examine the material and workmanship used in their construction, to witness the prescribed tests, and to see that this specification, as regards the boilers and work in connection, is conformed to in all respects by the contractors. The extent of supervision is described in the following paragraphs extracted from Admiralty instructions to overseers, and the contractors are to afford him every facility for their proper execution.

The plates and other material used in the construction of the boilers to be subjected to such tests as may be directed in the specification. Every plate used is to be carefully examined by the overseer for laminations, blisters, veins, and other defects, and to ensure that it is of the proper thickness and brand. No plate, angle, &c., which from any cause is considered by the overseer to be unfit for the intended use is to be fitted.

During the construction of the various parts of the boilers, the overseer is to satisfy himself that the dimensions as shown on the approved drawings are being adhered to by the contractors.

Whenever plates are flanged or welded, or in any case where iron or steel is worked in such manner that it is particularly liable to suffer in strength unless carefully handled, the overseer is to be present if possible on all occasions during the time the work on each article is in progress, and he is to fully satisfy himself that it is sound before he allows any part to be put in the boilers.

Samples of the rivets being used for the boilers are to be taken by the overseer during the progress of the work and tested as specified hereafter, and any batches of rivets found defective are to be rejected. Before rivets are put in, the overseer is to see that the plates are brought properly together, and that the holes are fair with one another. He is not to allow drifting on any account, but he is to see that they are carefully rimmed fair where necessary. He is also to make sure during the progress of the work that the rivets fill the holes completely, and that the heads are properly set up, well formed, and finished.

The overseer is to see that all internal parts of the boiler are riveted with rivets having heads and points of approved shape, and that any seams he considers necessary are riveted on the fire side. No snap heads are to be allowed in the internal parts. Any proposal for hydraulic riveting the internal parts is to be submitted to the Admiralty, with sketch of the proposed heads and points. In all parts where the rivets are not closed by hydraulic riveting machinery he is to see that the rivet holes are countersunk and that coned rivets are used. All holes in the plates, angles, &c., are to be drilled, and not punched, and are to be drilled in place after bending. The clearance between rivet hole and rivet before closing is not to be greater than approved by the overseer.

The overseer will see that the particulars of the form, dimensions, and pitch of the various stays shown on the drawings are adhered to, and samples of them are to be tested as directed in this specification; and he will be guided by his experience as a workman in testing and judging of the soundness of the forging and construction of the various stays.

He is to see that palm stays, if fitted, are forged from the solid and not welded, that all short stays are nutted on all flat surfaces except where otherwise approved and screwed to a pitch of twelve threads per inch for stays of 1 inch diameter and above, that the holes for the screwed stays in the water spaces are drilled and tapped together after the furnaces and combustion chambers have been riveted in place in the boiler, that the combustion chamber stays are drilled square to the bevel of the combustion chamber plates, and that no bevel washers are inside the chamber. Any girder stays used for combustion chambers are to be well bedded on to the tube plates to the satisfaction of the overseer.

The overseer is to see that the arrangement of the zinc plates shown on the approved drawings is adhered to, that the metallic surfaces in contact are filed bright, and that means are adopted to secure a firm grip of the clips by which the plates are attached.

The overseer is to witness the testing, in all cases, of the boiler tubes, in accordance with this specification, before they are put in the boiler.

When the boilers are reported to the overseer by the contractors as being completed, ready for testing by water pressure, the overseer is to witness a preliminary test of them in accordance with the specification, carefully observing with the assistance of gauges, and straight edges whether any bulging or deflection of the plates has taken place.

The official test will be conducted on all occasions in the presence of an inspecting officer. A test pressure gauge is supplied to the overseer from the Admiralty, and the official test is to be made with this gauge.

After the boilers have been tested by water pressure the overseer is to see that they are properly cleaned inside and outside, and then well painted with red lead. It is important that the whole surface of the boilers should be thoroughly cleansed of scale formed in manufacture before any paint is put on them. The boilers are not to be exposed to the weather till they are so painted, and properly cleaned and closed up to his satisfaction.

The overseer is to make himself fully acquainted with the progress of the whole of the work in its various stages, to satisfy himself that every part is sound before it is allowed to be put in the boilers, and to see that the following instructions for the treatment of mild steel are strictly complied with.

QUALITY OF AND TESTS FOR MATERIALS.

(a) *Steel generally.*

Quality of. All steel used throughout the whole of the work is to be of quality to be approved by the Admiralty. The names of the firms from whom it is proposed to obtain this material are to be submitted for approval before the order is placed. The quality and makers' names of all steel are to be marked on all drawings sent to the Admiralty.

Solid ingots for forgings. All steel forgings are to be made from solid ingots, and test pieces from each important completed forging will be subject to the tests for steel enumerated below, and facilities are to be afforded, by a proper system of numbering and registering the ingots, to enable the origin of any forging to be ascertained.

Testing in presence of Admiralty officers, &c. The whole of the temper and bending tests for furnace and other plates exposed to flame, also those for the important castings and forgings, and not less than 10 per cent. of all the other tests enumerated, are to be made in the presence and to the satisfaction of the Admiralty officers. These tests are intended to facilitate supply, but are not to preclude examination at the machinery contractors' works, and rejection of any plates or castings found defective after delivery. The tensile strength and elongation per cent. are to be stamped on each plate.

Copies of invoices to be supplied, &c. Copies of the invoices upon which the plates, bars, angles, rivets, stays, tubes, &c., for the boilers have been received, with the names of the manufacturers of the material and their tests, are to be supplied to the boiler overseer for his use in accepting them. When any orders are placed with steel makers, copies of the orders showing the particular purpose of the materials should be sent to the Admiralty overseer, with a copy of the specified tests.

(b) *Plate, Angle, and Bar Steel, for Boilers and any Steel Steam Pipes.*

Tests for steel plates, &c. Tensile test. Every plate, &c., used is to be tested as follows:—Strips cut lengthwise or crosswise are to have an ultimate tensile strength per square inch of section as follows:—(1) Plates, &c., not exposed to flame, not less than 27 tons, and not exceeding 30 tons; (2) lower front plates not less than 25 tons, and not exceeding 28 tons; (3) furnace plates not exceeding 25 tons, and not less than 23 tons; (4) fire-box and other plates exposed to flame not exceeding 26 tons, and not less than 24 tons. The elongation taken on a length of 8 inches, must be at least 20 per cent. for strips from shell and other plates which are not exposed to flame and which will not be flanged, and at least 25 per cent. for plates which are not exposed to flame and which will be flanged, at least 27 per cent. for strips from the furnaces, and at least 26 per cent. for strips from fire-box and other plates exposed to flame. Edge shearings, to bring plates to proper dimensions, are to be equal on opposite sides.

Temper tests. Strips cut lengthwise or crosswise, $1\frac{1}{2}$ inches wide, heated uniformly to a low cherry red, and cooled in water of 82° Fahrenheit, must stand bending double in a press to a curve of which the inner radius is one and a-half times the thickness of the steel tested.

The plates for furnaces and other parts exposed to flame are, in addition, to be tested by welding and forging, and some of the welded pieces are to be broken in the testing machine to ascertain the efficiency of the welding. Welding and forging tests of plates exposed to flame.

The pieces cut out for testing are all to be cut in a planing machine, to have the sharp edges removed, and are to be of parallel width for at least 8 inches of length. Test pieces.

The angle, tee, and bar steel are to stand such other forge tests, both hot and cold, as may be sufficient, in the opinion of the overseer, to prove the soundness of the material and fitness for the service intended. Forged tests of angle, &c., steel.

The furnaces are to be subjected to a hammering test if required before the machinery is accepted. The test is to be carried out in accordance with Admiralty practice. Hammering test.

(c) *Steel Rivets for Boilers.*

The whole of the rivets are to be properly heated in making, and care is to be taken that the finished rivets cool gradually. Manufacture.

The rivets are to be made from steel bars, the ultimate tensile strength of which should not be less than 25 tons, and should not exceed 27 tons per square inch, with a minimum elongation of 25 per cent. in a length of 8 inches. Tensile test

Pieces cut from the bars, heated uniformly to a low cherry red, and cooled in water of 82° Fahrenheit, must stand bending double in a press to a curve of which the inner diameter is equal to the diameter of the bar tested. Temper tests.

Samples from each batch of rivets are to be selected and are to stand the following forge tests satisfactorily without fracture:—The rivet to be bent double, cold, to a curve of which the inner diameter is equal to the diameter of the rivet tested; the rivet to be bent double when hot, and hammered till the two parts of the shank touch; the head to be flattened when hot, without cracking at the edges, till its diameter is two and a-half times the diameter of the shank. Finally, the shank of a rivet is to be nicked on one side, and bent over to show the quality of the metal. Forge tests.

(d) *Boiler Tubes.*

The tubes are to be made from acid or basic "open hearth" steel. Strips cut from the tubes must have a tensile strength not exceeding 25 tons and not less than 21 tons per square inch, with an extension of at least 25 per cent. in a length of 8 inches. Strips cut from the tubes flattened, heated to a blood heat and plunged into water 82° Fahrenheit temperature, should be capable of being doubled over a radius of $\frac{1}{2}$ inch without fracture. Pieces 2 inches long, cut from the ends of tubes under $\frac{3}{16}$ inch thick are to be capable, when cold, of being hammered down endwise until their length is reduced to $1\frac{1}{4}$ inch, and of being flattened until the sides Boiler tubes.

are close together in each case without fracture. The ends of the tubes under $\frac{3}{16}$ inch thick are to admit of being expanded cold by a roller expander, worked in a tube hole to an increase of diameter of 15 per cent., and hot by a solid drift to an increase of diameter of 20 per cent. Tubes $\frac{3}{16}$ inch and over in thickness are to admit of being expanded hot and cold to half the increases of diameter required for tubes under $\frac{3}{16}$ inch thick. The above tests are to be applied to 2 per cent. of the tubes, to be selected by the examining officers. The failure of the tubes selected to stand the specified tests in a satisfactory manner may render the whole of any delivery liable to rejection.

APPENDIX K.

German Government Rules for Boilers.

REGULATIONS RESPECTING STEAM BOILERS.

1. *Plating of Boilers.*—When the smallest dimensions of a cylindrical boiler exceeds 25 centimetres, or of a spherical-shaped boiler 30 centimetres, the portions of the boiler exposed to the flame, furnaces, and tubes shall *not* be made of cast iron.

Brass (copper) plates are only to be used for furnaces where their smallest dimension does not exceed 10 centimetres.

2. *Furnaces, Flues, and Combustion Chambers.*—Flues running round or through steam boilers must, at their highest point, be at least 10 centimetres below the minimum water level. This minimum level must, in lake and river vessels, be maintained when the vessel is inclined at an angle of 4°, and for sea-going ships at an angle of 8°. In boilers with a breadth of 1 to 2 metres, the distance to the water level must be at least 15 centimetres, and in boilers of greater breadth at least 25 centimetres.

These conditions do not apply to boilers in which the tubes are less than 10 centimetres diameter, nor to those wherein the portions of the plates in contact with steam are not liable to become red hot. The risk of plates becoming red hot may be assumed to be done away with if the heating surface exposed to the flame, before the flame reaches the point in question, is, with natural draught 20 times and with forced draught 40 times, greater than the area of the grate. The highest point in marine boiler flues, &c., is the *upper* surface of the plates in question.

3. *Feed Apparatus.*—Each boiler must have a feed valve which will close by boiler pressure when the feed is off. Where only one feed check-valve is provided shut-off valves must be fitted in the feed delivery pipes.

4. Each boiler must be fitted with two independent feed apparatus, each worked separately, and each to be capable, by itself, of keeping the boiler supplied. Several boilers connected together and used for the same purpose may, in this respect, be regarded as one boiler. Each feed apparatus to be capable of delivering 30 litres of water per square metre of heating surface per hour. Apparatus for boilers worked under forced draught, or having heating surface less than 35 times the grate area, must be proportionately larger. Each feed apparatus must have an independent suction pipe, or else a suitable arrangement of switch shut-off valves must be fitted. An auxiliary feed apparatus for a main boiler may be used for feeding a donkey boiler if there are independent delivery pipes, or if suitable switch shut-off valves are provided. If an injector is used for auxiliary feed purposes it must be capable of working satisfactorily with both high and low pressure steam; if it will not so work, some other provision must be made for use when pressure is low. Where a main boiler is used for driving winches in port, the main engine feed pumps will not be reckoned as a secondary feed apparatus. If supplementary feed arrangements, beyond those required by

the regulations, are provided, the piping for them must either be independent of that belonging to the regulation apparatus, or must be capable of being shut off from the latter by valves. Each main feed-pump must be fitted with an escape valve that cannot be shut off from the pump.

5. *Water Gauges.*—Each boiler must be fitted with a gauge glass, and also with a second means of ascertaining the height of the water. Each of these fittings must have a separate connection to the boiler, or if they have a common connection, the stand pipe must be at least 60 square centimetres in area, or 88 millimetres in diameter. Pipes for connecting water gauges to boilers must not be less than 16 centimetres in area, or, say, 45 millimetres in diameter. Long or much bent pipes must be larger in proportion. Internal connecting pipes must not be used, but two pipes may be led to one opening in the plating, provided the area of opening be made equal to the combined areas of the pipes. Gauge glasses are to be placed so as to be easily visible. The mark indicating lowest water level should not be higher than half-glass, and the mark showing highest point of heating surface should be above upper edge of lower nut which secures glass. The cocks must be easy to open and shut, and it must be possible to replace a broken glass without risk to the operator.

6. Water test cocks to be fitted : the lowest one must be placed at the level of the regulation water-line. All gauge cocks must be so fitted as to have a straight way through for cleaning them from scale or salt. Spindle valves must not be used.

7. *Water Level.*—The regulation lowest water level is to be distinctly marked on the gauge glass, and also in a prominent place on the boiler shell. The level of the flue, at its highest point, is also to be plainly and permanently marked on the shell of the boiler in way of the ship's beam.

Two water-gauge glasses are to be fitted on the boiler at the standard level of the water when the vessel is inclined as above, one on each side of the boiler, as far apart as possible, and symmetrically placed right and left of centre line of boiler.

When these two gauge glasses are fitted, the additional means named in Rule 5, for ascertaining the water level, may be dispensed with.

The marks indicating lowest water level are to be not less than 5 centimetres long, and are to be cut into the plating near to the water gauges. They are also to be distinguished by having the letters N. W. stamped on them. They must be on before the inspection of the boiler is made, and must be visible after it is lagged. Where water gauges of main boilers of river steamers and barges and of donkey boilers are direct on the boiler shell, a plate must be fitted at the lowest water-level mark with "Niedrigster Wasserstand" on it, and an arrow pointing to the glass. Where water gauges are connected to boilers by pipes (as must be the arrangements in all sea-going steamships) a similarly marked plate is to be fixed to the stand-pipe. The highest point of the heating surface is to be indicated as follows:—Where the water gauge is fitted direct to the boiler shell a plate indicating the point in question, and marked "Hochster Feuer berührter Punkt," is to be fixed near the gauge. Where the gauge is connected to the boiler by pipes a similar plate is to be affixed to the gauge standard. These plates are to be permanently secured—not with removable screws.

In cases where the fore and aft trim of the vessel is subject to considerable variation, the minimum water level (paragraph 2) must either be raised

10 centimetres, or the mark indicating highest point of heating surface must be raised a similar amount. The permanent mark showing regulation lowest water level must, however, correspond with the particulars given on the maker's name-plate, as hereafter specified.

A third water gauge must be fitted on every double-ended boiler.

8. *Safety-Valves.*—Each boiler must be fitted with, at the least, one reliable safety-valve.

When several boilers have a common steam chest, from which they cannot be separately disconnected, two safety-valves will be sufficient.

Boilers of steamships, locomotives, and portable engines must have at least two safety-valves.

In steamships, except those that are sea-going, one valve is to be placed in such a position that the load on it can easily be ascertained from the deck.

The valves must be so arranged that they can be readily lifted at any time. They are to be loaded so that they will blow off immediately the working pressure is reached.

Safety-valve areas must be such that after firing up for 15 minutes the pressure does not accumulate more than 10 per cent.

Provided that the springs are satisfactorily fitted and loaded, the area of safety-valves, in square millimetres, may be as follows for each square metre of heating surface :—

Pressure above atmosphere,	5	6	7	8	9	10	11	12	13	14	15	16
Area in square millimetres,	131	112	98	86	79	72	66	60	56	54	52	51

For boilers worked under forced draught, or in which the heating surface is less than 35 times the grate surface, the safety-valve areas must be correspondingly increased. Safety-valve boxes must be fitted with drain pipes that cannot be shut off. The waste steam pipes from all safety-valves must blow off into the open air. Blowing off into a tank, or into the chimney, will not be permitted. The thickness of the washers which determine the compressions of the safety-valve springs will be fixed by the Official Surveyor and recorded in his book, and no alterations must be made except by his authority. Exception is made, however (by the regulations of 19th December, 1883), in the case of long voyages, and the chief engineer of the vessel is permitted to re-adjust the washers in case the valves blow off too freely. To meet this case the washers may be fitted in halves.

9. *Pressure Gauge.*—Each boiler must have a reliable pressure gauge on which the highest regulation working pressure is to be plainly marked.

Boilers of steamships must have two pressure gauges—one clearly visible to the fireman, the other, except when the vessel is sea-going, in a convenient position for being observed from the deck. If a vessel has several boilers connected to a common steam chest, one pressure gauge on deck will be sufficient, in addition to the one on each boiler.

Gauges to be placed where they can easily be seen. Dials to be marked in kilogrammes per square centimetre (or in atmospheres,—1 atmosphere equals 1 kilogramme per square centimetre), and marking must go up to 50 per cent. above the regulation working pressure. In vessels having more than one boiler, one pressure gauge (in prominent position in the engine room), in addition to those on the boilers, will suffice. The pipe to this gauge must be connected to all boilers, but must be capable of being shut off from each.

10. *Name Plate on Boiler.*—(1) Each boiler must have the highest working pressure, the maker's name, the shop number of the boiler, and the date of completion, plainly and permanently stamped on it. Steamships, moreover, must also have the standard lowest water level marked on.

(2) Boilers that have already been constructed, at the date of these regulations, are not required to be altered to comply with them.

(3) The regulations for boilers of steamships apply in all cases in which boilers are permanently fixed in, or connected with, the vessel.

The address of the maker must be given as well as his name.

10*a*. These particulars must be stamped on a metal plate, which is to be fixed on the boiler by copper rivets, the heads of which are to be not less than 12 millimetres in diameter. The plate to be in such a position that it is easily visible when there are casings on the boiler.

11. *Testing of Boilers.*—Every new boiler must, on completion, be tested by water pressure.

The test pressure for boilers intended for a working pressure of not more than 5 atmospheres to be double the working pressure, and for boilers intended for a working pressure of more than 5 atmospheres, the test pressure is to be 5 atmospheres above the working pressure.

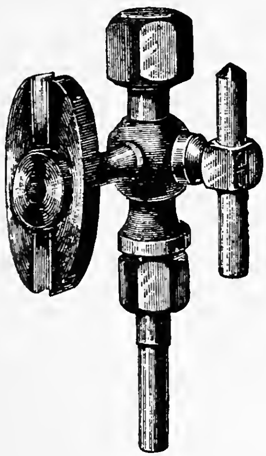


Fig. 339.

For boilers where the working pressure is under atmospheric pressure, the test pressure is to be 1 kilogramme per square centimetre. The plates of the boiler must withstand the test pressure without showing any permanent change of form, and must also remain tight under the pressure.

The boiler is to be regarded as leaky, if the water at the highest pressure leaks through in other forms than dewy moisture or pearly drops.

11*a*. When the boilers have been tested and found satisfactory, the copper rivets which fix the brass plate are to be officially stamped, and an impression of this mark is to be inserted on the certificate of test.

12. When boilers have undergone a thorough repair in the boiler shop, or when they have had to be stripped and extensively repaired on board, they must be tested by hydraulic pressure the same as new boilers.

When in boilers with internal furnace, the furnace has been taken out to repair or renew, and when in locomotives the fire-box has been taken out to repair or renew, or when in cylindrical boilers one or more plates have been renewed, the boilers must, on completion of such repairs, be tested by hydraulic pressure as above.

When a boiler has undergone considerable repairs the steam pressure to be allowed must be determined afresh.

13. *Test Gauge.*—The test pressure is to be registered by an open mercurial gauge, or by the standard pressure gauge provided by the inspecting official.

Each boiler to be tested must be fitted with a connection to suit the standard gauge, and by which the gauge may be applied by the inspecting officer.

To facilitate application of official test gauge, a flanged branch is to be provided immediately below the usual pressure gauge. The opening in the flange and cock must not be less than 7 millimetres, the length of the branch must be 60 millimetres, and its breadth 25 millimetres. Form to be as shown in sketch (fig. 336).

RULES FOR STRENGTH OF BOILERS.

5th August, 1890.

The quantities are expressed in centimetres and kilogrammes :—

(1 kilo. per sq. cm. = 1 atm. = 14·223 lbs. per sq. in English).

Formulæ for thickness of shell plate and working pressure :—

$$\text{Working pressure} = \frac{S \times R \times 2T}{D \times F}.$$

$$\text{Thickness of plate} = \frac{P \times D \times F}{S \times R \times 2}.$$

Where P = highest steam pressure (kilos. per sq. cm.).
S = lowest tensile strength of material.
T = thickness of shell plate.
D = greatest internal diameter of boiler.

$$F = \begin{cases} 4{\cdot}75 \text{ when rivet holes in longitudinal seams are drilled and rivets closed by hydraulic pressure.} \\ 4{\cdot}5 \text{ when longitudinal seams are double strapped and have holes drilled and rivets closed by hydraulic pressure.} \end{cases}$$

$$R = \frac{\text{Pitch} - \text{Diameter of rivet}}{\text{Pitch}}.$$

Or,

$$R = \frac{\text{Area of rivet} \times \text{No. of rivets in pitch} \times a}{\text{pitch} \times \text{thickness of plate}} \times 0{\cdot}9 \times \frac{S_1}{S}.$$

$$\text{Where } a = \begin{cases} 1 \text{ for lap and single riveted butt joints.} \\ 2 \text{ for double riveted, \&c., butt joints.} \end{cases}$$

S_1 = Tensile strength of rivet material. Unless actual test has been made, this is to be taken as 3300 for iron, and 3800 for steel; but it must not exceed 4800 kilos. per sq. cm. in any case.

The lower of these two values of R must be used.

S is to be taken as 3300 for iron; for steel the lowest value given by the tests is to be used, but this must not exceed 5000 kilos. per sq. cm., and the material must show an elongation of at least 20 per cent. in a length of 20 centimetres. Steel of higher tensile strength may only be used under specially arranged conditions as to tests, &c.

In riveted joints the shearing strength of the rivets must not be less than the tensile strength of the plate left between holes.

Butt-straps are to be of the same quality as the shell plates, and of not less than 75 per cent. of their thickness.

In boilers without a central circumferential seam the outside butt-strap must not be thinner than the shell plate. If the shell plates are thicker than 1·25 centimetres, the central circumferential seam must be double riveted, and if they are 2·5 centimetres or more it must be treble riveted.

Rivet holes must be at least one diameter of rivet from edge of plate. Diameter of rivet hole must not be less than thickness of plate, or more than 2 × thickness of plate, the former limit to apply to thick plates and the latter to thin ones.

Diagonal pitch must not be less than 2·1 × diameter of rivet.

Section 2.—Cylindrical Furnaces.

The thickness of plate for cylindrical furnaces is given by the formula—

$$\text{Thickness} = K\sqrt{P \times L \times D} + 0{\cdot}2 \text{ cm.}$$

Where P = working pressure (kilos. per sq. cm.).
L = length between stiffeners (centimetres).
D = external diameter of furnace (centimetres).
K = 0·0033 when the longitudinal joints are welded and fitted with double butt-straps, and the furnace is truly cylindrical.
K = 0·0035 when the longitudinal joints are lapped only.

The thickness of plate must not, in any case, be less than that given by the formula—

$$\text{Thickness} = \frac{P \times D}{C} + 0{\cdot}2 \text{ cm.}$$

Wherein C = 670 for plain furnaces without Adamson rings or other stiffeners.
C = 820 for furnaces with one Adamson ring, when the unsupported distance between the ring and the ends is not more than 122 cm.
C = 920 for furnaces with two Adamson rings, when the unsupported distance between the stiffeners is not more than 79 cm.
C = 1160 for Fox or Morison corrugated, or Purves ribbed furnaces. In corrugated furnaces D is the external diameter of the depressed corrugation, and in ribbed furnaces the external diameter of the plain portion between the ribs is to be taken.

The thickness proposed for any type of furnace, such as Farnley, Deighton, &c., must be referred for special consideration.

For the boilers of river steamers the constant addition may be 0·1 cm. in place of 0·2 cm.

Section 3.—Screw and Through Stays.

The strength of screw and through stays must not be taken higher than—

500 kilos. per sq. cm. for iron.
600 " " " steel.

Steel stays must not be welded.

In welded iron stays the section must be increased 20 per cent.

The pitch of screw stays in combustion chamber backs must not exceed 20 cm., and that of through stays in the steam space 45 cm.

Section 4.—Flat Surfaces.

The thicknesses of stayed flat surfaces are to be those given by the formula—

$$\text{Thickness} = 0{\cdot}15 \text{ cm.} + p\sqrt{\frac{P}{K}}.$$

Wherein p = greatest pitch of stays (in centimetres).
P = working pressure (kilos. per sq. cm.).
K = 2200 for stays screwed in and riveted over.
K = 2800 for stays screwed in and fitted with nuts.
K = 3000 for stays fitted with inside and outside nuts and washers, the outer washers being three times the diameter of the stay end and half the thickness of the plate.
K = 3400 for stays fitted with inside and outside nuts and washers, the outer washers having a diameter of ·6 × pitch, and a thickness of ·66 × thickness of plate, and being riveted to the plate.

If the combustion chamber top is supported by bridge stays only, which rest on the tube-plate, and are not assisted by any sling rods from the boiler shell, the thickness of the tube-plate must not be less than that given by the formula—

$$\text{Thickness} = \frac{P \times W \times p}{1800\,(p - d)}.$$

Wherein P = working pressure (kilos per square cm.).
W = width of chamber (tube-plate to back), in cm.
p = pitch of tubes, in cm.
d = internal diameter of plain tube, in cm.

If the tube-plates are of steel and are not exposed to the direct action of flame, the thickness may be reduced by $12\frac{1}{2}$ per cent.

Section 5.—Combustion Chamber Bridge Stays.

The strength of these stays is to be that given by the following formula:—

$$\text{Thickness} = \frac{P\,(L - p)\,S \times L}{K \times d^2}.$$

Wherein P = working pressure (kilos. per square cm.).
L = length of bridge stay, in cm.
p = pitch of stay bolts in bridge stay, in cm.
S = distance apart of bridge stays, in cm.
d = depth of bridge stay at centre, in cm.
K = 420 for one stay bolt in each bridge stay.
K = 630 for two or three stay bolts
K = 720 for four stay bolts.

"Thickness" stands for the collective thickness of the two plates of the bridge stay if constructed in that way.

If the plates are of steel they may be 10 per cent. thinner; and a further reduction may be made if they are assisted by slings from above.

GENERAL.

Manholes, &c., must be fitted with stiffening rings to compensate for plate cut away; they should not, as a rule, be less than 30 cm. × 40 cm., but in special cases 28 cm. × 38 cm. may be accepted. Cast-iron manhole covers cannot be accepted. It is strongly recommended that the studs of manhole covers be both screwed into the covers and riveted over on the inside.

Boilers are to be properly secured against movement both fore and aft and athwartships.

When there are several boilers they must be provided with proper shut-off valves to enable them to be worked independently.

Where the donkey boiler works at a lower pressure than the main boilers, satisfactory arrangements must be made to prevent steam passing from the latter into the former.

The space between the combustion chamber back and the back plate of the boiler should not be less than 12 cm., to allow for proper cleaning. In the cases of extra large boilers, and of boilers worked under forced draught, a greater space should be allowed.

Every marine boiler must be fitted with a blow-off valve and pipe, and also with a scum valve to draw the water from the level of the lower end of the water gauge. The blow-off pipe must be fitted with one cock on the boiler and a second on the ship's skin.

Wood must not be used either for lagging boilers or bulkheads in the neighbourhood of boilers.

All the larger mountings are to be secured to strong flanges riveted to the shell of the boiler; and the studs securing the mountings are not to penetrate the shell.

TESTS OF MATERIAL FOR STEEL BOILERS.

Regulations of 1st July, 1899.

All material to be of good and tough quality; any that proves brittle in working to be rejected.

A range of 5 kilos. will be allowed where ultimate tensile strength is 42 kilos. or under, and 6 kilos. where ultimate tensile is above 42 kilos.

Shell plates and other material not exposed to flame must show an elongation of at least 20 per cent. in a length of 200 millimetres; and material exposed to flame must show at least 22·5 per cent. on same length.

Material of less than 38 kilos. ultimate strength must elongate at least 25 per cent.

Bending tests are to be made on strips 25 millimetres wide, that have been heated to a dark red and quenched in water at 28° C.

They must stand bending without cracking to an angle of 180° over a radius, for shell plates, &c., of four times the thickness of the plate; and for material exposed to flame, of twice the thickness of the plate.

A tensile and a bending test is to be made from each shell plate. As regards other plates, a tensile test is to be made from every fourth plate; but, as far as possible, these test pieces should be so selected that there may be one from each furnace charge.

As regards plates other than shell plates and not exposed to flame, a bending test is to be made from each furnace charge. A bending test must be made from each plate that is exposed to flame.

Plates of over 41 kilos. tensile must be annealed if they have been worked in the fire.

Plates that are to be subjected to the direct action of flame should not have a higher ultimate tensile than 42 kilos. per square millimetre.

Steel of more than 46 kilos. ultimate tensile must not be used.

APPENDIX L.

Lloyd's Rules for Electric Light Installations on Board Vessels.

The following requirements as to the sizes, positions, and protection of the cables, and to the fitting of the cut-outs are now embodied as rules :—

LEADS OR CIRCUITS.

1. The sectional area of the copper wires in the cables should be at least in the proportion of 1 square inch per 1,000 amperes carried.

2. No single wire of greater size than 14 or of less than 18 standard wire gauge should be used. For portable leads, cables composed of stranded wires should be used having sufficient conductivity and flexibility for the purposes intended.

3. The copper used in all wires or cables should have a conductivity of at least 98 per cent. that of pure copper.

4. The insulation resistance of all wires, including portable leads, should be not less than 600 megohms per statute mile, after 24 hours' immersion in sea-water.

5. The insulating material used must not appreciably soften if subjected to a temperature of 180° F. If india-rubber insulation is used, the wires should be first covered with a layer of pure rubber, then with a separator, then with a layer of india-rubber coated tape. The whole should then be vulcanised together. The cable should afterwards be satisfactorily protected, preferably with a braided covering of waterproof fibre.

6. Wires which are insulated with any other material than india-rubber should fulfil the same conditions as to insulation resistance, and should be of equal durability with those above specified.

JOINTS.

7. Joints in branches, or of branches with leads of small circuits, must be made in properly constructed water-tight junction boxes, or should have the copper wires thoroughly soldered and the insulation carefully carried out, all the joints being made watertight. Joints in flow and return wires should not be made opposite one another. All joints should be in accessible positions, none being made in bunkers, cargo spaces, or spaces which may at any time be used for carrying cargo, stores, or baggage.

8. For soldering wires, resin only should be used as a flux.

9. Where practicable, the leads should be placed where they can be always accessible; if they are laid in wood battens the covers should be screwed on, not nailed, and care should be taken that the casings are so arranged that water will not lodge in them. Cables which are properly

covered with protective metal sheathing, or which are protected by galvanised wire armouring may be unencased. They should, however, be secured by screwed clips, not by staples. All sharp bends in cables should be avoided.

10. All cables which are liable to be exposed to the weather or moisture should be lead covered, or be otherwise specially protected. Where great heat is experienced, no wood casing should be used, but the cables should be protected by iron casings, or if they are not exposed to mechanical injury, they may be armoured with galvanised wire and fastened to decks or bulkheads with screwed clips spaced not more than 12 inches apart.

11. If cables are led through cargo spaces, coal bunkers, or spaces which may at any time be used for carrying cargo, stores, or baggage, or which are not at all times accessible, they should be strongly protected against damage, preferably by iron casings. If they are led through metal tubes, these must be strongly secured, and should be fitted so that the water cannot lodge in them.

Armoured cables may be used without casings or tubes provided they are strongly secured to the under side of decks or to bulkheads by screwed clips, and provided they are armoured in conformity with the standard of the Engineering Standards Committee, viz.:—

For cables below $\frac{1}{2}$ inch diameter over lead by galvanised steel wires ·072 diameter, for cables $\frac{1}{2}$ inch to 1 inch over lead by two layers of steel tape, each ·03 inch thick, for cables above 1 inch to 2 inches diameter by two layers of steel tape, each ·04 inch thick, and for larger cables by two layers of steel tape ·06 thick.

12. Where cables pass through beams, bulkheads, or other iron work, they should be led through special fittings of sheet lead, hard wood, or vulcanised fibre to prevent their being chafed, and where they pass through decks they should be led through metal tubes lined with wood, or vulcanised fibre, and securely fastened to the decks, standing at such a height above the deck level that water cannot stand above them. Where cables pass through water-tight bulkheads the fittings should be provided with brass water-tight screwed glands.

13. In vessels having spaces allotted alternately for passengers and cargo, the lamp fittings in these spaces should be removable, and the terminals so arranged that they can be properly covered up with strong metal covers, or the whole of the fittings should be similarly provided with strong metal covers. The main switches and cut-outs should be outside these spaces, or if placed inside, they should be in strong iron boxes provided with iron covers, or otherwise securely arranged to prevent the fittings being tampered with.

DISTRIBUTION.

14. A main switchboard should be fitted in the dynamo-room, to which all the main circuits throughout the ship should be brought, a switch and cut-out being fitted thereon for each circuit. The auxiliary switchboards for further sub-division of the current should be placed in conveniently accessible positions, and each such switchboard should be similarly fitted with a separate switch and cut-out for each sub-circuit. Cut-outs should be fitted to each lamp circuit where these are made with reduced size of wire.

If vessels are wired on the double-wire system cut-outs should be fitted to each cable of these circuits.

15. In cases where electric lights are used for the mast-head light and side lights, the switches controlling these lights should be placed in a position where they can be controlled by the officer of the watch, or other responsible person, and cannot be tampered with by other members of the crew, or by passengers, etc.

16. The switchboards should be of slate or other incombustible material. The switches should be on the quick-break principle, and should be so constructed that they must be either full "on" or completely "off"—that is, they must not be able to remain in an intermediate position. They should have ample rubbing surfaces and their conductivity should not be less than that of the wires connected to them.

17. Cut-outs should be fitted to each main or auxiliary circuit, on the switchboards, as near as possible to the switches of these circuits. If the switchboard is not fitted near the dynamo, or if more than one dynamo may be used on any one circuit, then cut-outs should also be fitted to the main cable as near as possible to each of the dynamo terminals.

18. All other cut-outs should also be in easily accessible places, and as near as possible to the commencement of the cables or wires they protect. They should be mounted on slate or other incombustible bases, and be arranged so that the fused metal may not be a source of danger, and where fitted with covers these should be incombustible.

19. All fuses should be of easily fusible and non-oxidisable metal, and should be so proportioned as to melt with a current 100 per cent. in excess of the normal current; that is, they should melt with a current in the proportion of 2,000 amperes per square inch of section of the wires they protect. The fuses for branch wires to single lights should, however, if of tin wire, be of not greater size than 22 S.W.G.

20. The fuses for each cable should be made of standard dimensions, so that a large fuse cannot be used for a small cable by mistake, or, if wire fuses are used, permanent instructions should be fitted on or near each switchboard, giving particulars of the proper size of fuse for each circuit.

21. In shaft passages and in damp places, all lamp switches and cut-outs should be of a strong water-tight pattern, or should be placed in water-tight boxes having hinged or portable water-tight covers. No switches or cut-outs are to be placed in bunkers.

22. There should be no joints in the cables leading from the dynamo to the main switchboard, nor in those leading from the main to auxiliary switchboards, nor should branches to single lamps be taken off these cables.

23. A voltmeter should be supplied with each installation. If more than one dynamo is fitted, neither being capable of the whole of the output, an ampere meter should be supplied with each dynamo.

JOINTS W TH HULL.

24. In vessels fitted on the single-wire system, all the joints with the hull should be placed in accessible positions. Those for single lamps or for small cables should be made with brass screws not less than three-eighth of an inch in diameter, carefully tapped into the iron or steel, having white

brass washers, between the wires and the vessel, or the wires should be soldered to brass-faced washers. For larger cables and for the pole of dynamo the cable wires should be properly sweated into brass or copper shoes, which should be bolted to the vessel. The iron or steel where contact is made should be filed bright, and the area of contact should not be less than eight times the section of the copper of the cable.

IN VESSELS CARRYING PETROLEUM.

25. The single-wire system must not be adopted for any part of the installation. Switches and cut-outs must not be fitted in places liable to the accumulation of petroleum vapour or gas, and all lamps in places where it is possible for gas to accumulate must be made with an outer glass globe made air-tight. All wires in such places are to be lead covered, or the insulation of the cables employed is to be of such a nature as not to be affected by petroleum. No joints of cables, switches, or cut-outs should be fitted in the pump-room, but the wires for each lamp therein should be carried to the lamp from a distributing junction box placed outside the pump-room or companion.

The following paragraphs referring to the effect of the electric light installations upon the compasses are issued as suggestions, not as rules:—

POSITION OF DYNAMOS AND OF ELECTRIC MOTORS.

26. The position and type of dynamos and electric motors should be such that the compasses will not be affected. Dynamos and large motors should be at least 30 feet from the standard compass.

CABLES.

27. In vessels fitted with continuous-current dynamos, and wired on the single-wire system, no single cable should be carried within 15 feet of any compass, and cables conveying heavy currents should be fixed at still greater distance. If it is necessary to fix any cables within this distance, then for all parts of the vessel lighted from this cable the concentric or double wire system should be adopted, the return wire being carried as near the flow as possible in the vicinity of the compasses.

London, *17th June*, 1909.

APPENDIX M.

Refrigerating Machinery and Appliances.

LLOYD'S REGISTERS' RULES.

On the application of the owners of vessels fitted for carrying refrigerated cargoes, the Committee will authorise their surveyors to survey the refrigerating machinery and appliances, and in those cases where the following conditions are complied with and a satisfactory report is received from the surveyor, certificates of these surveys will be issued and the notation "R.M.C." (*in red*) (*i.e.*, Refrigerating Machinery Certificate) will be made against the vessel's name in the Society's Register Book, and in the special list of vessels fitted with refrigerating appliances. In cases in which the refrigerating machinery and appliances are constructed under the special survey of the Society's Surveyors and to their entire satisfaction, the notation "+ R.M.C." (*in red*) will be made in the Register Book. The name of the maker and description and number of refrigerating machines, whether single or duplex, and the refrigerating power of the machines will be recorded in the special list in the Register Book, as will also the number and capacity of insulated cargo chambers and the nature of insulation and the method employed for cooling the holds.

1. The insulation must be sound and in good order and of efficient construction. The details of construction showing the amount and nature of the insulating material employed in the various parts are to be reported to the Committee.

Bilge suction and sounding pipes and ballast tank air and sounding pipes, passing through insulated spaces, should be well insulated to prevent their being frozen up. No sluice valves, scuppers, or drain pipes are to be fitted, which will permit drainage from spaces outside of the insulated chambers into the bilges of the insulated holds.

It is *recommended* that the woodwork of the insulation over tunnel tops be fastened with screws to facilitate the examination of this part, and that extra strong battens of American elm be fitted upon it under the hatches. Insulated removable portions are to be arranged in the bulkhead insulation, where required, to give easy access to sluice valves and bilge suction roses. The bottoms, sides, and coamings of all insulated hatches and limbers should be painted to prevent decay.

Thermometer tube flanges and covers should be arranged so that water does not run down and freeze in them when taking the temperature.

Cargo battens should be provided for the floor or deck and the sides of the chambers previous to loading the homeward cargo. Those for the sides of the chambers should be fastened, and should be at least $1\frac{1}{2}$ inches in depth, and 2 inches wide, one batten being placed over each frame or ground, the others being intermediately arranged. The battens for the floor and decks should be at least 2 inches × 2 inches.

Where the brine system of refrigerating is employed, the brine circulating pipes and tanks should not be galvanised on the inside.

In cases where internally galvanised tanks and cooling pipes have been fitted, the brine cooling and return tanks, if closed, should be provided with two ventilating pipes communicating with the atmosphere. If the tanks are not closed, the cooling room should be efficiently ventilated.

2. The refrigerating machinery is to be of approved construction, and of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running 18 hours per day. For cargo capacities of above 70,000 cubic feet the machinery is to be either duplex or in duplicate.

3. A sufficient amount of spare gear is to be supplied and stowed where it is readily accessible.

No spare gear will, however, be required in cases where two complete sets of refrigerating machines are fitted, *each* being of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running 18 hours per day, provided all the working parts of these machines are interchangeable.

When two similar machines are fitted, each connected to different cargo compartments, one set of spare gear suitable for either machine will suffice.

Where one single dry air machine is fitted to each compartment, the following will be required :—

1 crank-shaft with eccentric sheaves, complete, or one half shaft if the halves are interchangeable.
1 piston-rod and nuts for steam and air cylinders.
1 set of piston-rod and connecting-rod brasses.
1 piston, complete, for each steam and air cylinder.
1 cylinder cover for each pattern used in steam and air cylinders.
1 air-pump bucket and rod.
1 circulating-pump bucket and rod.
1 pair main bearing brasses, complete.
Main and cut-off valves for each steam cylinder.
Balance springs and rings for steam and air slide valves.
False valve face for each pattern fitted in steam cylinders, with screws.
1 eccentric rod for each pattern used.
1 eccentric strap for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders, for each pattern used.
2 main bearing bolts.
1 set of connecting-rod and piston-rod bolts.
Full set of air valves and seats for air compressor.
1 set of inlet and outlet valves and 1 set valve faces (if fitted) for air-expansion cylinders, with screws.
1 set of valves for air, circulating, and feed pumps.
1 set of escape valve springs.
50 suction springs.
50 delivery springs.

50 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler.
6 tubes for air-drying chamber.
Assorted bolts, studs, and nuts.
1 set of lead-lined nuts for air-expansion cylinder cover.
A quantity of packings and joint rings.

Where one duplex or two single dry air machines are fitted to each compartment the following will be required :—

1 crank-shaft with eccentric sheaves complete, or one half shaft if the halves are interchangeable.
1 piston-rod and nuts for steam and air cylinders.
1 set of connecting-rod and crosshead brasses.
1 piston for H.P. steam cylinder.
1 piston, complete, for air compressor; and 1 for air-expansion cylinder.
1 set of piston springs for each steam cylinder.
1 cylinder cover for each pattern used in air compression and expansion cylinders.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves and spindles with nuts complete for H.P. steam cylinder.
Balance springs and rings for steam and air slide valves.
1 H.P. steam cylinder valve and valve face with screws.
1 eccentric sheave, strap, and rod for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders for each pattern used.
2 main bearing bolts.
1 set of connecting-rod and piston-rod bolts.
Half set of air valves and seats for air compressor.
1 inlet and 1 outlet valve, and half set of valve faces (if fitted) for air expansion cylinder, with screws.
1 set of valves for air, circulating, and feed pumps.
1 set of escape valve springs.
20 suction springs.
40 delivery springs.
40 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler and 6 for air-drying chamber.
Assorted bolts, studs, and nuts.
$\frac{1}{2}$ set of lead-lined nuts for air-expansion cylinder cover.
A quantity of packings and joint rings.

Where one single ammonia or carbonic anhydride compression machine is fitted :—

1 crank-shaft with eccentric sheaves, complete, or one half shaft if the halves are interchangeable.
Piston and rods complete with nuts for each steam cylinder and gas compressor.

1 air pump bucket and rod.
1 circulating pump bucket and rod.
1 pair main bearing brasses, complete.
1 set of connecting-rod and crosshead brasses.
Main and cut-off valves for steam cylinders.
1 valve spindle for each pattern used and nuts complete.
1 eccentric strap and rod for each pattern used.
1 brine pump complete.
1 cover for each pattern used.
2 main bearing bolts.
1 set of connecting-rod and piston-rod bolts.
1 set compressor suction and delivery valves with springs and boxes, complete.
1 set of valves for air, circulating, feed, and brine pumps.
Crank-shaft for fan engine.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting-rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
Quantity of leather packings and joint rings.

For ammonia and carbonic anhydride compression machines, the following spare gear will be required, where one duplex or two single machines are fitted to each compartment:—

1 crank-shaft, or one half shaft if the halves are interchangeable.
1 steam piston-rod and nut for each pattern used.
1 piston for H.P. steam cylinder, with springs, complete.
1 set of piston rings for each steam cylinder.
1 set of piston rings for each size of compressor.
1 compressor piston-rod and nuts, complete, for each pattern used.
1 air-pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves for H.P. steam cylinder.
Main and cut-off valve spindles and nuts for H.P. steam cylinder.
1 eccentric sheave, strap, and rod, for each pattern used.
1 brine pump complete.
1 cover for each end of gas compressor, except where screwed plugs are used.
2 main bearing bolts.
$\frac{1}{2}$ set of connecting-rod and piston-rod bolts.
$\frac{1}{2}$ set compressor suction and 1 delivery valve with springs and box, complete.

1 set of valves for air, circulating, feed, and brine pumps.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting-rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks and leather for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
A quantity of joint rings.

In cases where an independent circulating water pump is used, and its work *cannot* be performed by the main or auxiliary engines, a duplicate pump complete should be fitted.

In cases where an independent circulating water pump is used, and its work *can* be performed by the main or auxiliary engines, a pump bucket and rod should be carried, and ½ set of valves for water end.

In cases where an independent surface condenser with air, circulating and feed pumps combined is fitted, and its work *cannot* be performed by the main engines:—

1 crank-shaft with eccentric sheaves complete.
1 piston and rod complete for each pattern used.
1 eccentric strap and rod complete for each pattern used.
1 slide valve and spindle complete for each pattern used.
1 pump bucket and rod complete for each pattern used.
1 set of connecting-rod and piston-rod bolts and nuts.
1 set of valves for air, circulating and feed pumps.
6 condenser tubes and 12 ferrules.

In cases where an independent surface condenser with air, circulating, and feed pumps combined is fitted, and its work *can* be performed by the main engines:—

1 pump bucket and rod complete for each pattern used.
1 set of connecting-rod and piston-rod bolts and nuts.
½ set of valves for air, circulating, and feed pumps.
6 condenser tubes and 12 ferrules.

PERIODICAL SURVEYS.

9. The complete periodical survey required in par. 4 will consist of the following:—

The insulation throughout the holds is to be carefully examined and tested for dryness and fulness by sounding with a hammer and by boring. The test holes are to be afterwards efficiently closed. Special attention is to be paid to the spaces under the snow boxes, trunks, and hatches where dampness may accumulate, to the sides under stringers and under decks and to the tunnel tops. All limber hatches are to be removed, the limbers cleared, and the suction pipes and roses, sluices, and sounding pipes are to be examined. Hatches, air trunk-ways, and thermometer tubes with their connections and fastenings are to be examined, and where trunk-ways pass through water-tight bulkheads, the water-tight doors are to be examined and worked.

The trunk-ways should be as air-tight as practicable, and their fastenings should be secure.

The steam pipes, water pipes, and connections, the crank-shaft and bearings, connecting-rods, steam and air cylinders, pistons, slides and valves, compressors and pistons, compressor rods and glands, surface condenser, and air or gas coolers, circulating, air, feed, and bilge pumps, are to be carefully examined, and the condensers and coolers tested if deemed necessary.

The auxiliary machinery, where fitted, is also to be examined.

The spare gear is to be examined.

In dry air machines special attention is to be given to the condition of the air expansion cylinders, their pistons and valves. In other machines special attention is to be given to the condition of the compressors, including the pistons, rods, and glands, and to the expansion valve.

The refrigerator coils and their connections and the brine pipes and tanks, where fitted, are to be carefully examined and tested if deemed necessary.

Where the brine may escape to the bilges, the cement is to be examined.

The machinery is to be examined under working conditions and tested on the snow box or refrigerators, the time and fall of temperature being noted.

APPENDIX N.

Lloyd's Rules for Vessels Fitted for Burning and Carrying Liquid Fuel.

(*Section* 49, *Steel Ship Rules.*)

1. In vessels fitted for burning liquid fuel, the record "fitted for liquid fuel" will be made in the Register Book.

2. The compartments for carrying oil fuel must be strengthened to withstand efficiently the pressure of the oil when only partly filled and in a seaway. They must be tested by a head of water extending to the highest point of the filling pipes or 12 feet above the load line, or 12 feet above the highest point of the compartment, whichever of these is the greater.

3. If peak tanks or other deep tanks are used for carrying liquid fuel the riveting of these should be as required in the case of vessels carrying petroleum in bulk. The strengthening of these compartments must be to the committee's satisfaction.

4. Each compartment must be fitted with an air pipe to be always open discharging above the upper deck.

5. Efficient means must be provided by wells and sparring or lining to prevent any leakage from any of the oil compartments from coming into contact with cargo or into the ordinary engine-room bilges.

6. If double bottoms under holds are used for carrying liquid fuel, the ceiling must be laid on transverse battens, leading at least two inches air space between the ceiling and tank top and permitting free drainage from the tank top into the limbers.

7. The pumping arrangements of the oil fuel compartments and their wells must be absolutely distinct from those of other parts of the vessel, and must be submitted for approval.

If it is intended to carry sometimes oil and sometimes water ballast in the various compartments of the double bottom, the valves controlling the connections between these compartments and the ballast donkey pump, and also those controlling the suctions of the special oil pump, must be so arranged that the suction for each separate compartment cannot be connected at the same time to both pumps.

8. No wood fittings or bearers are to be fitted in the stokehold spaces.

9. Where oil fuel compartments are at the sides of, or above, or below the boilers, special insulation is to be fitted where necessary to protect them from the heat from the boilers, their smoke boxes, casings, etc.

10. If the fuel is sprayed by steam, means are to be provided to make up for the fresh water used for this purpose.

11. If the oil fuel is heated by a steam coil the condensed water should not be taken directly to the condenser, but should be led into a tank or an open funnel mouth, and thence led to the hot-well or feed tank.

12. The above arrangements are applicable only to the case of oil fuel, the flash point of which, as determined by Abel's close test, does not fall below 150° Fahrenheit.

London, 17*th June*, 1909.

APPENDIX O.

Board of Trade Rules for Safety-valves.

Provisions of the Act as regards Safety-valves.—Every steamship of which a survey is required by the Act shall be provided with a safety-valve upon each boiler, so constructed as to be out of the control of the engineer when the steam is up, and, if such valve is in addition to the ordinary valve, it shall be so constructed as to have an area not less, and a pressure not greater, than the area of and pressure on that valve.

The surveyors are instructed that, in all *new boilers* and whenever *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible and be cast in one with the chest.

If any person place an undue weight on the safety-valve of *any* steamship, or, in the case of steamships surveyed under the Act, increase such weight beyond the limits fixed by the engineer-surveyor, he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

The engineer-surveyor shall declare, amongst other things, the limits of the weight to be placed on the safety-valves; that the safety-valves are such, and in such condition, as required by the Act; and that the machinery is sufficient for the service for the time he fixes, and is in good condition.

Area of Safety-valves.—When natural draught is used, the area per square foot of fire-grate surface of the locked-up safety-valves should not be less than that given in the following table opposite the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels and to vessels which have not previously received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the table, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

When forced draught is used, the area of the safety-valves should not be less than that found by the following formula:—

$$\frac{A \times C}{20} = \text{area of valves required.}$$

Where A = area of valves as found from the table.
C = estimated coal consumption in lbs. per sq. ft. of grate per hour.

TABLE CXIII.—Safety-valve Areas.

(natural draught.)

Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.	Boiler Pressure.	Area of Valve per Square Foot of Fire-grate.
Lbs.	Sq. inch.	Lbs.	Sq. inch.	Lbs.	Sq. inch.	Lbs.	Sq. inch.
15	1·250	67	·457	119	·279	170	·202
16	1·209	68	·451	120	·277	171	·201
17	1·171	69	·446	121	·275	172	·200
18	1·136	70	·441	122	·273	173	·199
19	1·102	71	·436	123	·271	174	·198
20	1·071	72	·431	124	·269	175	·197
21	1·041	73	·426	125	·267	176	·196
22	1·013	74	·421	126	·265	177	·195
23	·986	75	·416	127	·264	178	·194
24	·961	76	·412	128	·262	179	·193
25	·937	77	·407	129	·260	180	·192
26	·914	78	·403	130	·258	181	·191
27	·892	79	·398	131	·256	182	·190
28	·872	80	·394	132	·255	183	·189
29	·852	81	·390	133	·253	184	·188
30	·833	82	·386	134	·251	185	·187
31	·815	83	·382	135	·250	186	·186
32	·797	84	·378	136	·248	187	·185
33	·781	85	·375	137	·246	180	·184
34	·765	86	·371	138	·245	189	·183
35	·750	87	·367	139	·243	190	·182
36	·735	88	·364	140	·241	191	·181
37	·721	89	·360	141	·240	192	·181
38	·707	90	·357	142	·238	193	·180
39	·694	91	·353	143	·237	194	·179
40	·681	92	·350	144	·235	195	·178
41	·669	93	·347	145	·234	196	·177
42	·657	94	·344	146	·232	197	·176
43	·646	95	·340	147	·231	198	·176
44	·635	96	·337	148	·230	199	·175
45	·625	97	·334	149	·228	200	·174
46	·614	98	·331	150	·227	201	·173
47	·604	99	·328	151	·225	202	·173
48	·595	100	·326	152	·224	203	·172
49	·585	101	·323	153	·223	204	·171
50	·576	102	·320	154	·221	205	·170
51	·568	103	·317	155	·220	206	·169
52	·559	104	·315	156	·219	207	·169
53	·551	105	·312	157	·218	208	·168
54	·543	106	·309	158	·216	209	·167
55	·535	107	·307	159	·215	210	·166
56	·528	108	·304	160	·214	211	·166
57	·520	109	·302	161	·213	212	·165
58	·513	110	·300	162	·211	213	·164
59	·506	111	·297	163	·210	214	·164
60	·500	112	·295	164	·209	215	·163
61	·493	113	·292	165	·208	216	·162
62	·487	114	·290	166	·207	217	·161
63	·480	115	·288	167	·206	218	·161
64	·474	116	·286	168	·204	219	·160
65	·468	117	·284	169	·203	220	·159
66	·462	118	·281				

N.B.—When oil fired or with forced draught, take an assumed grate area equal to the total heating surface divided by 30.

When the pressure exceeds 180 lbs. per square inch, the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves, and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

In ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

Examination of Safety-valves.—The surveyor, in his examination of the machinery and boilers, is particularly to direct his attention to the safety-valves, and, whenever he considers it necessary, he is to satisfy himself as to the pressure on the boiler by actual trial.

The surveyor is to examine the whole of the valves, weights, and springs at every survey.

The responsibility of seeing to the efficiency of the mode by which the valves are fitted, so as to be out of the control of the engineer when steam is up, rests with the surveyor, who should see that the method adopted is efficient and approved of by the Board of Trade.

The safety-valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should, in all cases, be arranged so that it can be worked by hand either from the engine-room or stokehole.

Care should be taken that the safety-valves have a lift equal to at least one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste steam pipe, should each have an area not less than the area of valves required by Table cxiii.; that each valve-box has a drain-pipe fitted at its lowest part. In the case of lever valves, if the holes in the lever are not bushed with brass, the pins must be of brass; iron and iron working together must not be passed. The valve seats should be secured by studs and nuts.

Surveyor to See Valves Weighted.—When the surveyor has determined the amount of pressure, he is to see the valves weighted accordingly, and the weights or springs fixed in such a manner as to preclude the possibility of their shifting or in any way increasing the pressure. The limit of the weight on the valves is to be inserted in the declaration, and should it at any time come to a surveyor's knowledge that the weights have been shifted or the loading of the valves otherwise altered, or that the valves have been in any way interfered with, so as to increase the pressure, without the sanction of the Board of Trade, he is at once to report the facts to the Board of Trade.

Spring Safety-valves.—If the following conditions are complied with, the surveyor need raise no question as to the substitution of spring loaded valves for dead-weighted valves:—

(1) That at least two valves are fitted to each boiler.
(2) That the valves are of the proper size, as required by Table lxxvi.
(3) That the springs and valves be so cased in that they cannot be tampered with.
(4) That provision be made to prevent the valves flying off in case of the springs breaking.

(5) That the requisite safety-valve area is cased and locked up in the usual manner of the Government valves.
(6) That screw lifting gear be provided to meet all the valves.
(7) That the size of the steel of which the springs are made is to be found by the following formula:—

$$\sqrt[3]{\frac{s \times D}{c}} = d.$$

s = the load on the spring in lbs.
D = the diameter of the spring (from centre to centre of wire) in inches.
d = the diameter, or side of square, of the wire in inches.
c = 8000 for round steel.
c = 11,000 for square steel.

(8) That the springs be protected from the steam and impurities issuing from the valves.
(9) That, when valves are loaded by direct springs, the compressing screws abut against metal stops or washers when the loads sanctioned by the surveyor are on the valves.
(10) That the springs have a sufficient number of coils to allow a compression under the working load of at least one-quarter the diameter of the valve.

The size of steel of springs of safety-valves should not, as a rule, be less than $\frac{1}{4}$ inch.

In no case is the surveyor to give a declaration for spring-loaded valves unless he has tried them under full steam and full firing for at least twenty minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. In special cases, or when the valves are of novel design, the results of the test under full steam should be reported to the Board, but, if the surveyor is fully satisfied with them, he need not delay the granting of the declaration for the vessel pending the approval of the Board. If, however, the accumulation of pressure exceed 10 per cent. of the loaded pressure, he should withhold his declaration, and report the case to the Board of Trade, accompanied by a sketch, if necessary, and stating the strength pressure and the working pressure of the boilers.

In the case of safety-valves, of which the principle and details have already been passed by the Board of Trade, the surveyor need not require plans to be submitted, so long as the details are unaltered, of which he must fully satisfy himself; but in any new arrangement of valves, or in any case in which any detail of approved valves is altered, he should, before assuming the responsibility of passing them, report particulars, with a drawing to scale, to the Board of Trade.

The tracings of new safety-valve designs should, if possible, be transmitted to the Board of Trade for consideration before the construction of the safety-valves is commenced.

Owners, Masters, and Engineers to see that Valves are kept in Proper Order.—It is clearly the duty of the masters and engineers of vessels to see, in the intervals between the surveys, that the locked-up safety-valves, as well as the other safety-valves and the rest of the machinery, are in proper working order. There is no provision in the Merchant Shipping

Act, 1854, exempting the owner of any vessel, on the ground that she has been surveyed by the Board of Trade surveyors, from any liability, civil or criminal, to which he would otherwise be subject. The Act of Parliament requires the Government safety-valve to be out of the control of the engineer when the steam is up; this enactment, far from implying that he is not to have access to it, and to see to its working, at proper intervals when the vessel is in port, rather implies the contrary; and the master should take care that the engineer has access to them for that purpose. Substantial locks that cannot be easily tampered with, and as far as possible weather-proof, should be used for locking up the safety-valve boxes.

All Tests for Pressure and Accumulation are to be made with Board of Trade Gauges.—In witnessing the hydraulic tests of boilers, &c., and in witnessing all safety-valve tests for accumulation of pressure, the surveyors are to use the pressure-gauges supplied by the Board of Trade for the purpose. No steam gauge should be used without having a syphon filled with water between it and the boiler.

APPENDIX P.

Bureau Veritas Rules for Boiler Safety-Valves and Metal Tests, etc.

$$\text{Area of safety-valves} = \frac{8 \cdot 7}{\sqrt[4]{(P - 18)}} \text{ square inches.}$$

P is the working pressure in pounds per square inch.
No valve should be less than $1\frac{1}{4}$ inches diameter.

Thickness of shell plate is deduced from the following :—

$$\text{Working pressure} = \frac{\alpha \, 2 \, R}{D} (t - 0 \cdot 04).$$

Therefore,
$$t = \frac{P \, D}{\alpha \, 2 \, R} + 0 \cdot 04 \text{ inch.}$$

D is the greatest inside diameter in inches.
t is the thickness in inches.
0·04 the allowance for corrosion.
R is the tensile stress in lbs. per square inch (which is the ultimate ÷ 4), and must be stated by the maker.
α is the ratio of the plate between the holes to the full plate, or pitch − diameter of rivet ÷ pitch.

Tests of Materials.

Steel Plates.—Tensile in tons plus percentage of elongation in 8 inches not less than 50.

Iron Plates.—Tensile with 23 and 18 per cent. in 8 inches; across, $21\frac{1}{2}$ and 12 per cent. in 8 inches.

Bend cold, $\frac{10}{12}$ 32nds, 120° with 100° across.
,, $\frac{17}{20}$ 32nds, 90° ,, 70° ,,

Nickel Steel, 3·25 Ni 0·35 *C.*—$\frac{37}{42}$ ton and 20 per cent. in 4.

Copper Plates.—Tensile, $12\frac{1}{2}$ and 22 per cent. in 8 inches. Bend double at dull red heat.

Copper Tubes.—Tensile, 13 and 30 per cent. in 4 inches.

Brass Plates.—Tensile, 19 with × 12 per cent.

Steel Pin Rivets.—$\frac{24}{29}$ ton over $\frac{23}{32}$, 25 per cent. under $\frac{13}{32}$nds, 20 per cent. in 8 inches.

Brass Tubes.—Internal pressure, 430 lbs. per square inch. Condenser tubes, 570 lbs. per square inch.

Brass Castings.—$11\frac{1}{2}$ tons, 8 per cent. in 4 inches.

Iron Castings.—Minimum pressure, $1\frac{5}{8}$ square, $6\frac{1}{4}$ in supports, series shocks. Pressure of 27 lbs. weight falling 12 inches up to 18 inches by 2 inches increase.

Common castings, $7\frac{1}{2}$ tons, and fall 10 to 16 inches.

APPENDIX Q.

Admiralty Rules for Testing Materials for Machinery.

(e) *Steel Castings for Machinery.*

Steel castings. Tests

Steel castings for the machinery are to satisfy the following conditions:—Tensile strength, not less than 28 tons per square inch, with an extension of 2 inches of length of at least 23 per cent. Bars of the same metal, 1 inch square, should be capable of bending cold, without fracture, over a radius not greater than $1\frac{5}{8}$ inches, through an angle depending on the ultimate tensile strength, this angle to be not less than 90° at 28 tons ultimate strength, and not less than 60° at 35 tons ultimate strength, and in proportion for strengths between these limits. For intricate thin castings the extension in 2 inches of length is to be at least 10 per cent., and the bending angle is to be not less than 20° at 28 tons ultimate strength, and 15° at 35 tons ultimate strength, and in proportion for strengths between these limits. Test pieces are to be taken from each casting. All steel castings are also to satisfactorily stand a falling test, the articles being dropped from a height of 12 feet (or as may be approved) on a hard macadamised road or a floor of equivalent hardness.

It is to be distinctly understood that contractions or defects in steel castings are not to be made good by patching, burning, or by electric welding, without the sanction of the Admiralty overseers.

(f) *Steel Forgings for Machinery.*

Steel forgings.

All steel forgings are to satisfy the following conditions:— Ultimate tensile strength, not less than 28 tons per square inch, with an extension in 2 inches of length of at least 30 per cent. Bars of the same metal, 1 inch square, should be capable of being bent cold, without fracture, through an angle of 180° over a radius not greater than $\frac{1}{4}$ inch. Test pieces are to be taken from each forging. Crank and propeller shafts are to have test pieces taken from each end, and the ultimate tensile strength of the material of these shafts must not exceed 32 tons per square inch.

For all important forgings, such as crank and propeller shafts, connecting- and piston-rods, the forgings are to be gradually and uniformly forged from solid ingots, from which at least 30 per cent. of the top end of the ingot has been removed before forging, and at least 3 per cent. of the total weight of the ingot from the bottom end after forging. The sectional area of the body of the finished forging is to be not more than $\frac{1}{6}$ the original sectional area of the ingot.

(*g*) *Cast Iron.*

Cast iron.

Test pieces to be taken from such castings as may be considered necessary by the inspecting officer. The minimum tensile strength to be 9 tons per square inch, taken on a length of not less than 2 inches. The transverse breaking load for a bar 1 inch square, loaded at the middle between supports 1 foot apart, is not to be less than 2000 lbs.

(*h*) *Gun-metal, Naval Brass, and White Metal.*

Composition, analysis, and tests of gun-metal and naval brass, &c.

The gun-metal used for all castings throughout the whole of the work supplied by the contractors is, unless otherwise specified, to contain not less than 8 per cent. of tin, and not more than 5 per cent. of zinc, the remainder to be of approved quality copper. The exact proportion of tin above 8 per cent. being arranged as may be required, depending on the use for which the gun-metal is intended. The ultimate tensile strength of gun-metal is to be not less than 14 tons per square inch, with an extension in 2 inches of length of at least 7½ per cent. The composition of any naval brass used is to be :—Copper, 62 per cent. ; zinc, 37 per cent. ; and tin, 1 per cent. All naval brass bars are to be cleaned and straightened. They are to be capable of (1) being hammered hot to a fine point, (2) being bent cold through an angle of 75° over a radius equal to the diameter or thickness of the bars. The ultimate tensile strength of naval brass bars ¾ inch diameter and under is not to be less than 26 tons per square inch, and for round bars above ¾ inch diameter, and square bars not less than 22 tons per square inch, whether turned down in the middle or not. The extension in 2 or 4 inches of length is to be at least 10 per cent. Breaks within less than ½ inch of the grip are not to count. Cuttings from the propellers and other important gun-metal castings and naval brass work will be sent to Portsmouth dockyard for analysis. The white metal used for bearing surfaces is to contain at least 85 per cent. of tin, not less than 8 per cent. of antimony, and about 5 per cent. of copper; zinc or lead should not be used. The brasses are to be carefully tinned before filling with white metal.

Castings and forgings

The gun-metal castings are to be perfectly sound, clean, and free from blowholes. The steel castings are required to be clean, sound, and to be out of twist, and as free as possible from blowholes ; the steel forgings are required to be quite sound, clean, and free from all flaws. All castings and forgings must admit of being machined, planed, and bored to the required dimensions ; and no piecing, patching, bushing, stopping, or lining, will be permitted, nor will any manufactures in which these conditions have been infringed be accepted. In cases of doubt as to the suitability of castings or other materials, for the purpose intended, early reference should be made to the inspecting officer to avoid subsequent delay by the rejection of such parts after delivery. The whole of the steel plates, angles, and rivets used in the construction of any part of the work supplied is to be manufactured by the Siemens-

Martin process. All castings are to be of steel or gun-metal, except where otherwise specified; and all forgings, plates, bolts, &c., are to be of steel. No steel is to be toughened without sanction from the Admiralty.

Copper for Pipes.

Copper. Strips cut from the steam and other pipes, either longitudinally or transversely, are to have an ultimate tensile strength of not less than 13 tons per square inch when annealed in water, with an elongation in length of 2 inches or 4 inches of not less than 35 and 30 per cent. respectively. Such strips are also to stand bending through 180° cold until the two sides meet, and of hammering to a fine edge without cracking.

Treatment of Mild Steel.

Hammer scale to be removed. All plates and furnaces for boilers and steam pipes are to be treated as follows, with a view of removing the black oxide or scale formed during manufacture:—The plates and furnaces, previous to their being taken in hand for working, are to stand for not less than 8 hours in a liquid consisting of 19 parts of water and 1 of hydrochloric acid. The plates should be placed in the bath on edge, and not laid flat. When the plates and furnaces are removed from the dilute acid both the surfaces are to be well brushed and washed to remove any scale which may still adhere to them. They should then be placed in another similar bath filled and kept well supplied with fresh water, or be thoroughly washed with a hose, as may be found necessary. The plates on removal from the fresh water should be placed on edge to dry. This treatment is to be carried out on the premises where the boilers and pipes are made.

Cold bending. All plates or bars which can be bent *cold* are to be so treated; and if the whole length cannot be bent cold, heating is to be had recourse to over as little length as possible.

Hydraulic flanging. The front and end plates of the boilers, and also all other plates, including those for the combustion chambers, *are to be flanged by hydraulic pressure,* and in as few heats as possible.

Dangerous temperature to be avoided. In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a "blue heat"—say from 600° to 400° Fahrenheit. Should this limit be reached during working, the plates or bars should be reheated.

Annealing. Plates or bars which have been worked while hot are to be subsequently annealed simultaneously over the whole of each plate or bar.

Failure of metal. In cases where any bar or plate shows signs of failure or fracture in working it is to be rejected. Any doubtful cases are to be referred to the Admiralty.

APPENDIX R.

RECOMMENDATIONS OF THE BRITISH MARINE ENGINEERING DESIGN AND CONSTRUCTION COMMITTEE FOR MAIN CYLINDRICAL BOILERS SUMMARISED.

The Steel used must be made in an open-hearth furnace, either by the acid or basic process.

Ordinary shell plates and longitudinal stays shall have a tensile strength from 28 to 32 tons per square inch.

Special steel plates and stays may be as high as 35 tons tensile. If *higher tensile* than 35 tons is desired, special arrangements must be made with the Registering Body.

Steel plates for welding and flanging, and the bars for screw stays and for rivets, 26 to 30 tons tensile.

Steel strips for welded tubes must not have a higher tensile than 28 tons.

Wrought-iron bars may be used for screwed stays and allowed the same working stress as for steel, if it is tested and stands 21·5 tons tensile with an extension of 25 per cent. in 8 inches.

The methods of testing and expected results are to be as prescribed by the British Engineering Standards Committee.

The hydraulic test to be applied to new boilers shall be twice the working pressure when that does not exceed 100 lbs. per square inch; when it does, then the test pressure is to be 50 lbs. in excess of one and a half times the W.P.

For boilers which have been on service, the test must not be more than 1·5 the W.P.

Cylindrical shells.—The W.P. shall equal $\dfrac{(t-2) \times S \times J}{C \times D}$.

Where t is the thickness of the shell plate in 32nds of an inch.
- S is the minimum tensile strength of the plates in tons per square inch.
- J is the percentage of strength of joint as found in the usual way (*v.* p. 906).
- C is a coefficient which is 2·67 when the longitudinal seams are with double butt straps, and 2·87 when they are lapped.
- D is the inside diameter of the outer drum of plating.

The maximum pitch of rivets in inches $= (K \times t) + 1{\cdot}625$, where K is as given on p. 911, but with no restriction to 10 inches.

For furnaces the following are the rules:—

(i.) *Plain.*

$$\left.\begin{aligned} \text{W.P.} &= \frac{1{,}450\,(t-1)^2}{(L+24) \times D}, \\ \text{W.P.} &= \frac{50}{D}\,\{10\,(t-1) - L\}. \end{aligned}\right\} \text{The minimum to be taken.}$$

Where D is the external diameter in inches.
t is the thickness in 32nds of an inch.
L is the length in inches between the lines of substantial support from other parts.

(ii.) *Corrugated.* $\text{W.P.} = C(t-1) \div D.$

Where D is the least external diameter as taken at the bottom of corrugations.
t is the thickness in 32nds of an inch.
C is 480 for the Fox, Morison, Purves & Deighton, and similar furnaces; 510 for Leeds Forge suspension furnace.

No furnace should be more than $\frac{26}{32}$ inch thick.

Flat plates supported by screwed stays.

(*a*) $$\text{W.P.} = (t-1)^2 \times C \div a^2 + b^2.$$

Where t is the thickness in 32nds of an inch.
a is the distance apart of the rows of stays in inches.
b is the pitch of the stays in the rows in inches.
$C = 50$ when the stays are riveted over only.
$C = 55$ for screwed stay tubes when expanded in three holes.
$C = 75$ when screwed stays are fitted with nuts on the outer ends and exposed to flame, and 88 when not exposed.
$C = 100$ when the stay simply passes through the plate and fitted with nuts inside and outside the plate.

When the stays pass through the plates and have nuts inside and outside with substantial washers under the outer nuts, 3·5 times the stay in diameter, and the thickness t_w is not less than 0·2 the diameter of the stay, then

(*b*) $$\text{W.P.} = 100\{(t-1)^2 + 0{\cdot}15\,t_w^2\} \div a^2 + b^2,$$

when the washers are in diameter at least 0·67 the pitch of the stays, and in thickness t_w is not less than 0·6 the thickness of the plate and riveted to it.

(*c*) $$\text{W.P.} = 100\{(t-1)^2 + 0{\cdot}35\,t_w^2\} \div a^2 + b^2,$$

when the plate is stiffened by strips riveted to it whose breadth is at least 0·67 the pitch, and the thickness t_w is not less than 0·6 the thickness of the plate to which it is efficiently riveted.

(*d*) $$\text{W.P.} = 100\{(t-1)^2 + 0{\cdot}55\,t_w^2\} \div a^2 + b^2,$$

when there is a doubling plate at least 0·6 its thickness efficiently riveted to it. Then

(*e*) $$\text{W.P.} = 100\{(t-1)^2 + 0{\cdot}85\,t_w^2\} \div a^2 + b^2.$$

For front tube plates having spaces between the nests of tubes, and between them and the boiler shell, and between them and the furnaces and longitudinal stays. t_w is the thickness when a doubling plate is riveted to the tube plate.

(*a*) $$\text{W.P.} = 55\{(t-1)^2 + 0{\cdot}55\,{}_w^2\} \div a^2 + b^2,$$

when all the stay tubes at margins are screwed into plates, but without nuts.

(*b*) $\text{W.P.} = 72\,\{(t-1)^2 + 0{\cdot}55\,t_w^2\} \div a^2 + b^2$, when all have nuts.

(*c*) When only alternate stay tubes have nuts, the factor is 63 instead of 72.

For back tube plates, where there must be no nuts,

(*d*) $\text{W.P.} = 38\,(t-1)^2 \div p^2$,

where p is the mean pitch got by adding together the four sides of the quadrilateral from tube stay centre to tube stay centre and dividing by 4.

(*e*) For the W.P. for the front tube plate for stays other than the margin ones the same rule applies, and if the stay tubes have nuts the factor is 49.

Screw stays to combustion chambers must have nine threads per inch, and whether of steel or tested wrought iron

$$\text{W.P.} = (d - 0{\cdot}267)^2 \times 8{,}250 \div a \times b,$$

where d is the diameter over thread, a and b are the vertical and horizontal pitches in inches.

Longitudinal stays must have six threads per inch.

$$\text{W.P.} = \frac{(d - 0{\cdot}34)^2}{a \times b} \times \frac{S}{28},$$

where d is the diameter over thread and S the minimum tensile strength of bars in tons.

Stay tubes, whether of lapwelded wrought iron or steel, shall be allowed a working stress of 7,500 lbs. per square inch at bottom of threads.

Dished ends shall have a radius of curvature not exceeding the diameter of the shell if without a stay. The inside radius of curvature at flange shall be not less than four times the thickness.

$$\text{W.P.} = 15 \times S\,(t-1) \div R,$$

where t is the thickness in 32nds of an inch, R the radius of curvature, and S the minimum tensile strength of steel in tons.

Mountings and Fittings.—Water-gauge glasses may be $\frac{3}{4}$ or $\frac{5}{8}$ inch in outside diameter, and 20, 18, 16, or 14 inches long. Stand pillars for water gauges 2·5 inches diameter inside. Single-ended boilers over 15·5 feet diameter must have two, and double-ended over 16 feet three such gauges. Two independent means of feeding and origin of feed.

Safety valves to be such that there is not on any one valve a greater load than 2,600 lbs. The aggregate area on any one boiler shall be found thus—

$$\text{Rule for area in sq. inches} = \text{total heating surface in sq. feet} \times \left(\frac{1{\cdot}25}{p + 15}\right),$$

p being the working or gauge pressure in lbs. per square inch.

The valves may be loaded to $2\frac{1}{2}$ per cent., but not to more than 5 lbs. in excess of the working pressure.

The waste steam pipe and passages from the valves must have a sectional area equal to 0·01 square inch for each square foot of total heating surface.

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